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REAR ADMIRAL JOHN HALLIGAN, JR., U. S. NAVY.

On October 29, 1925, Capt. John Halligan, Jr., U. S. Navy, was appointed Engineer-in-Chief of the Navy and Chief of the Bureau of Engineering, with the rank of Rear Admiral. Admiral Halligan was born in South Boston, Mass., in 1876 and entered the Naval Academy, Annapolis, in 1894, graduating in 1898 at the top of his class. During the Spanish-American War he had active service on the U. S. S. *Brooklyn*, being on board during the battle of Santiago on July 3, 1898. Admiral Halligan was promoted to the rank of ensign on April 4, 1900; to the rank of lieutenant (J.G.) on April 4, 1903; to the rank of lieutenant on January 1, 1904; to the rank of lieutenant commander in March, 1910; to commander in August, 1916; was temporarily commissioned a captain on July 1, 1918, and received the permanent rank of captain on

June 3, 1921. During the World War, Admiral Halligan served as aide on the staff of the commander of the patrol forces in the Atlantic Fleet, and later as chief of staff to the commander of the United States Naval Forces in Europe.

Admiral Halligan is an officer of wide engineering experience, both afloat and ashore. In 1902 and 1903, while in command of the U. S. T. B. *Rodgers,* he, with his crew, was assigned to assist the Liquid Fuel Board in its extensive tests with fuel oil. His services in this connection were highly praised by the Board. He was senior engineer officer of the U. S. S. *Nebraska* from November 25, 1908, to June 8, 1909, and served in the design division of the Bureau of Engineering from June 12, 1909, to September, 1912. During this latter detail he made a special study of turbine machinery in England. He served as fleet engineer of the Atlantic Fleet from September, 1914, to September, 1915, at which time he was detailed as head of Engineering Post-graduate School at Annapolis, Md., serving in that capacity until April, 1917. From November, 1920, to May, 1923, he was in charge of the Naval Engineering Experiment Station at Annapolis.

While in command of the U. S. S. *Ohio* in 1919 and 1920, Admiral Halligan carried on extensive radio experiments, being also in charge of the *Iowa* which was fitted with radio control and used as a target. His last command was the U. S. S. *Detroit,* from July, 1923, to June, 1925. When selected as Chief of the Bureau of Engineering, Admiral Halligan was attending a course of instruction at the Naval War College, Newport, R. I.

HIGH TEMPERATURE INSULATION.

By Lieut.-Comdr. C. S. Gillette, U. S. Navy, Member.

The modern trend toward the use of higher steam pressures and temperatures has necessitated the development of more exact information in connection with the use of all the ordinary engineering materials, and has aroused interest in some of the newer materials not so well known, but which are particularly suited to the changed conditions. The behavior of metals at elevated temperatures offers a vast field for research—a field which for the first time is being thoroughly investigated. So, also, heat insulating materials must be reconsidered from the standpoint of the new conditions, and in so doing much information has developed which may be used to improve the old standard installations as well as to insure the efficiency of the new.

A brief survey of the existing information on heat insulation in engineering literature will convince almost anyone that in spite of its apparent simplicity, the whole matter is exceedingly involved; the information is misleading, no conclusion is susceptible of positive proof, and the answer to the questions of what kind and how much heat insulation to use for a specific purpose must be pretty much accepted on faith. However, laboratory tests can establish certain facts, costly experience can determine others, and common sense the remainder, so that in the end the situation is not as bad as it might at first appear.

It is generally accepted from an insulation point of view that any surface, the temperature of which is 100 degrees F. or more, may be considered as heated, and if the heat radiated

therefrom constitutes an economic loss the surface should be insulated. Surfaces below such a temperature and above about 50 degrees F. should often be insulated also, but here the heat transfer is usually to, not from, the surface, and the heat insulation is applied for a different purpose, viz., to prevent "sweating" or condensation of moisture from the warmer surrounding atmosphere and consequent inconvenience or damage from dripping water. Surfaces below 50 degrees F. may usually be considered as "cold," or refrigerated if the conductance of heat thereto from the surrounding atmosphere results in an economic loss, and hence should be well insulated also.

The advantages of proper heat insulation of piping and machinery are not limited to savings in operating expense through conservation of fuel or power, but, by preventing dissipation of heat to the surrounding atmosphere, the comforts of life on shipboard, and thereby the operating efficiency of the personnel, are more easily maintained. In machinery spaces particularly, good insulation insures suitable working conditions without which the personnel cannot be expected to produce the best results; under battle conditions, lack of insulation might render an already uncomfortable situation so much worse as to have a very serious affect on the outcome of the conflict.

Considering heat insulation as that used for temperatures of 100 degrees F. or more, the type and quantity to be used should be the cheapest both in first cost of material and in the labor of application which will serve the purpose and at the same time have a reasonably satisfactory service life. From this point of view it is, therefore, most convenient to consider the question with reference to the zones or ranges of temperature commonly encountered in service. The temperature ranges most convenient for this purpose are more or less commonly accepted as follows:

(*a*) 100 degrees F. to 119 degrees F. (28 inches vacuum — 26.5 inches vacuum).

(*b*) 120 degrees F. to 267 degrees F. (26.5 inches vacuum — 25 pounds gauge).

(*c*) 268 degrees F. to 338 degrees F. (25 pounds gauge — 100 pounds gauge).

(*d*) 339 degrees F. to 388 degrees F. (100 pounds gauge — 200 pounds gauge).

(*e*) 389 degrees F. to 500 degrees F. (200 pounds gauge — 300 pounds gauge plus superheat).

(*f*) 501 degrees F. to 750 degrees F. (300 pounds gauge and above plus superheat).

(*g*) 751 degrees F. to 1500 degrees F. (Boiler tube casings).

(*h*) 1501 degrees F. to 2400 degrees F. (Boiler furnace casings).

In so far as shipboard applications are concerned, the temperature ranges from (*a*) to (*f*) inclusive, cover most of the piping and machinery using steam in present and probable future designs. Temperature ranges (*g*) and (*h*) are ordinarily encountered only in boiler settings.

Heat insulating materials are usually supplied in one or more of the following forms:

(1) Sectional pipe covering; formed in two sections in 36 inch lengths to fit pipes up to and including 10 inch diameters, in standard thickness, double standard thickness, and other thicknesses as required.

(2) Segmental pipe covering; formed in curved segmental blocks with edges beveled to fit pipes above 10 inches in diameter and in thickness as required.

(3) Block covering; formed in flat rectangular blocks 6 by 18 or 36 inches for use on flat or large cylindrical surfaces, and in thicknesses as required.

(4) Brick covering; formed in standard brick sizes, 9 by 4-1/2 by 2-1/2 inches.

(5) Plastic covering; insulating material in powdered form mixed with some suitable bonding material, usually asbestos fiber, to be mixed with water and applied by hand.

(6) Cement covering; powdered insulating material mixed with a bonding medium, to be mixed with water and applied by hand as a jointing for insulating brick or other forms.

(7) Granular covering; a coarse granular or lump form of insulation for use in filling into fully enclosed spaces, or for mixing with Portland cement and water to form an insulating concrete.

(8) Felted or quilted covering; this form is used where the insulating material is of a more or less flexible nature, such as hair or wool, and the various fibrous materials, including asbestos.

Of the commonly used materials, "85 per cent magnesia," that is, hydrated magnesium carbonate bonded with asbestos, is the most widely known and may be considered as the best insulation for use within its field of application. The material is standardized, is available in all parts of the country, and is supplied by several manufacturers in the sectional, segmental, block, and plastic forms. It has a very low thermal conductivity and is otherwise very satisfactory as an insulation, except that its resistance to abrasion is low and it must, therefore, be protected by suitable lagging of heavy canvas or sheet steel in practically all cases. It is supplied commercially as sectional pipe covering in certain standardized thicknesses dependent on the pipe diameters; for convenient reference the standard thicknesses are given in the following table:

Pipe diameter (I.P.S.)	Standard thickness	Double standard thickness
1/2 to 1-1/2 inches, inclusive	7/8	1-3/4
2 to 3-1/2 inches, inclusive	1-1/32	2-1/16
4 to 6 inches, inclusive	1-1/8	2-1/4
7 to 10 inches, inclusive	1-1/4	2-1/2

The maximum temperature for which "85 per cent magnesia" should be used, however, is 500 degrees F. It has been claimed by many that this material may be used with satisfactory results up to temperatures of 600 degrees F. and more, but the best practice seems to favor the more conservative limit. If no more satisfactory materials were available, it might be necessary to use the material at temperatures in excess of its calcination point and doubtless fair service might reasonably be expected for a limited length of time, but when more satisfactory materials are available no good reason can be seen for approving "85 per cent magnesia" for use at temperatures at which its efficient service life is admittedly doubtful at best. However, its low thermal conductivity renders it entirely satisfactory and desirable when used as an outer layer over an inner layer of insulating material with higher heat resistance but higher thermal conductivity; in such case the inner layer of insulating material with greater heat resistance must be of sufficient thickness to insure that the temperature gradient has dropped below the 500 degrees F. point at its outer surface over which the "85 per cent magnesia" layer is applied.

Hair felt, another commonly known material, is used in some of the lower temperature ranges. This material has excellent insulating qualities and stands up well in service if properly applied within the temperature range for which it is suitable. The temperature limit for hair felt does not seem to have been very well established but it is generally considered good practice to use this material at temperatures well below 200 degrees F. The material has a tendency to rot if subjected to moist conditions in service; of course, almost any heat insulation loses practically all of its insulating value when wet, and proper installation consists in the greatest possible assurance against the entrance of moisture.

For the temperature range (*a*), viz., 100 degrees F. to 119 degrees F., either a 1 inch thickness of hair felt, or a 1 inch

or standard thick layer of "85 per cent magnesia" is considered the best practice. Difference in cost of material or application will usually dictate which material to use; hair felt, in general, being considered the most satisfactory where exposure to abrasion or vibration in service is likely to be severe.

For temperature range (*b*), viz., 120 degrees F. to 267 degrees F., a 1 inch or a standard thickness of "85 per cent magnesia" is most commonly used.

For temperature range (*c*), viz., 268 degrees F. to 338 degrees F., a 1-1/2 inch thickness of "85 per cent magnesia" is considered good practice.

For temperature range (*d*), viz., 339 degrees F. to 388 degrees F., a double standard thickness of "85 per cent magnesia" should be used.

For temperature range (*e*), viz., 389 degrees F. to 500 degrees F., a 3 inch thickness of "85 per cent magnesia" is the best practice; ordinarily this thickness should be applied in two 1-1/2 inch layers.

For temperature range (*f*), viz., 501 degrees F. to 750 degrees F., the insulation thickness should be 3-1/2 inches thick and applied in one or more layers of heat resistant insulation, or with an inner layer of heat resistant insulation at least 1-1/2 inches thick and an outer layer of "85 per cent magnesia" of about 2 inches in thickness.

It must be understood that the foregoing practice is ordinarily required only for the larger size pipes, 4 inch diameter or more, and other large machinery surfaces. For pipes between 2 inches and 4 inches in diameter the practice should be somewhat modified; in general, it is advisable to require only standard thickness of "85 per cent magnesia" up to and including the 339 degrees F. to 388 degrees F. temperature range, double standard thickness of "85 per cent magnesia" for temperature range 389 degrees F. to 500 degrees F., and a 3 inch insulation thickness for the highest temperature range. For pipes below 2 inches in diameter further reduction of

insulation is again advisable; the standard thickness of "85 per cent magnesia" should be carried up to and including the 389 degrees F. to 500 degrees F. temperature range, and a 2 inch insulation thickness used for the highest temperature range.

We thus come to the field of high temperature insulation, viz., from 500 degrees F. to the highest temperature commonly reached in service, approximately 2400 degrees F. This is a field of comparatively recent development and for the present the importance of one form or another of diatomaceous earth material in this field overshadows about everything else, in much the same fashion that magnesium carbonate does in the lower temperature ranges. Before going further with a description of the various forms and applications of this insulation, it may be of interest to go into some detail in connection with the interesting basic material itself, a material about which comparatively little is commonly known.

Diatomaceous earth consists of the siliceous residue from minute vegetable diatoms that lived millions of years ago. The diatom gets its name from the ancient Greek word "diatomeo" meaning to cut in two, indicating their reproduction by division. This is true for the most part; the most common method of reproduction being for the two valves of a diatom to separate at the girdle, each valve developing later into a complete frustule. A complete diatom consists of an upper and lower valve (fitting together like the halves of a circular pill box) connected by a hoop or girdle.

Diatoms may be examined either as living organisms (for they are found in almost all water—fresh or salt—running or quiet) or what is perhaps more interesting, by making slides from small particles of diatomaceous earth which consists of fossil diatoms. These minute plant organisms have been fittingly referred to as "Jewels of the Sea" and when one has a good field of them on a slide under the microscope with the light shining through just right they are certainly deserving of such a title.

The minuteness of these diatoms will be realized from the fact that one cubic foot of raw material contains more than 40,000,000,000 individual cells, each cell the remains of a single organism. Each cellular unit is a hollow siliceous "skeleton," all organic matter being leeched out during countless centuries. A magnification of 40 diameters will in most cases serve to show the great variety of shape and structure and a magnification of from 400 to 600 diameters brings out the detail workings in all their delicate tracery; see Figures 1 and 2.

Diatoms should not be confused with infusoria. The former is entirely a primitive form of plant life (bacillariae) whereas the latter is an animal organism. Both diatoms and infusoria grow in much the same manner and live under similar conditions.

There are two general types of diatoms. One type inhabits the shores and shallow bottoms of the oceans, lakes and streams. The second class are free floating and move about on or near the surface of oceans or lakes and are known as Plankton diatoms. As the individual diatoms die they sink to the bottom and in the course of immense periods of time large accumulations are piled up.

One very large deposit of diatomaceous earth from which practically all the material now being used commercially is obtained, is in Santa Barbara County, California, near the town of Lompoc. This particular deposit was laid down during the Miocene era and was probably formed about two million years ago.*

The deposit ranges in depth from around 800 feet to 1400 feet in thickness, thick layers of diatomaceous silica being interlaid with relatively thin layers of clay or other sediment, indicating that the accumulation of diatoms was interrupted from time to time either by geologic changes or by changes in ocean currents.

* "A Miocene Catastrophe" by David Starr Jordon, Natural History, Volume I, page 18.

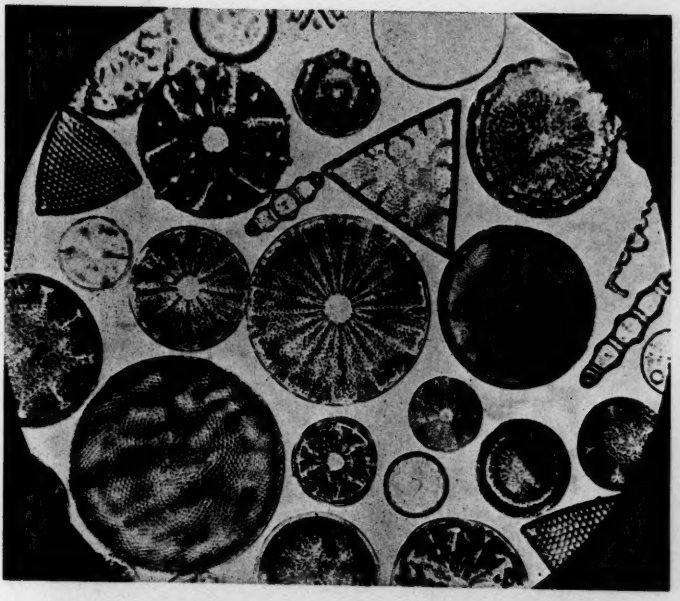


FIG. 1.—REPRESENTATIVE GROUP OF ENLARGED DIATOMS.

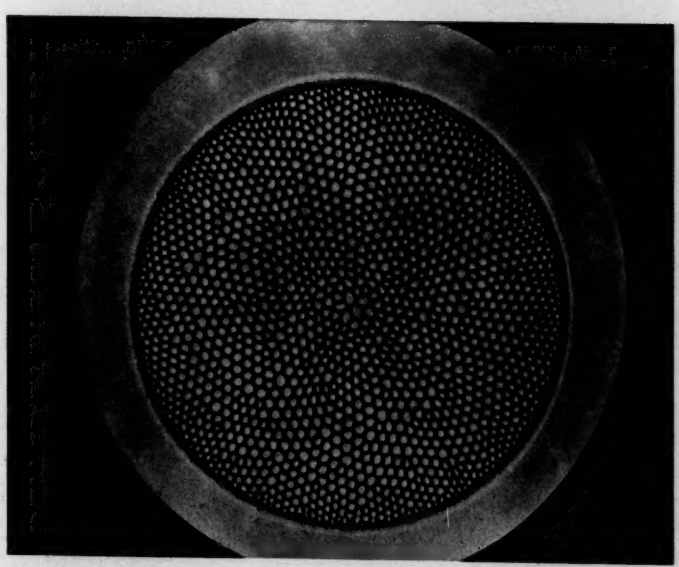


FIG. 2.—INDIVIDUAL DIATOM GREATLY ENLARGED (COSCINODISCUS RADIATUS).

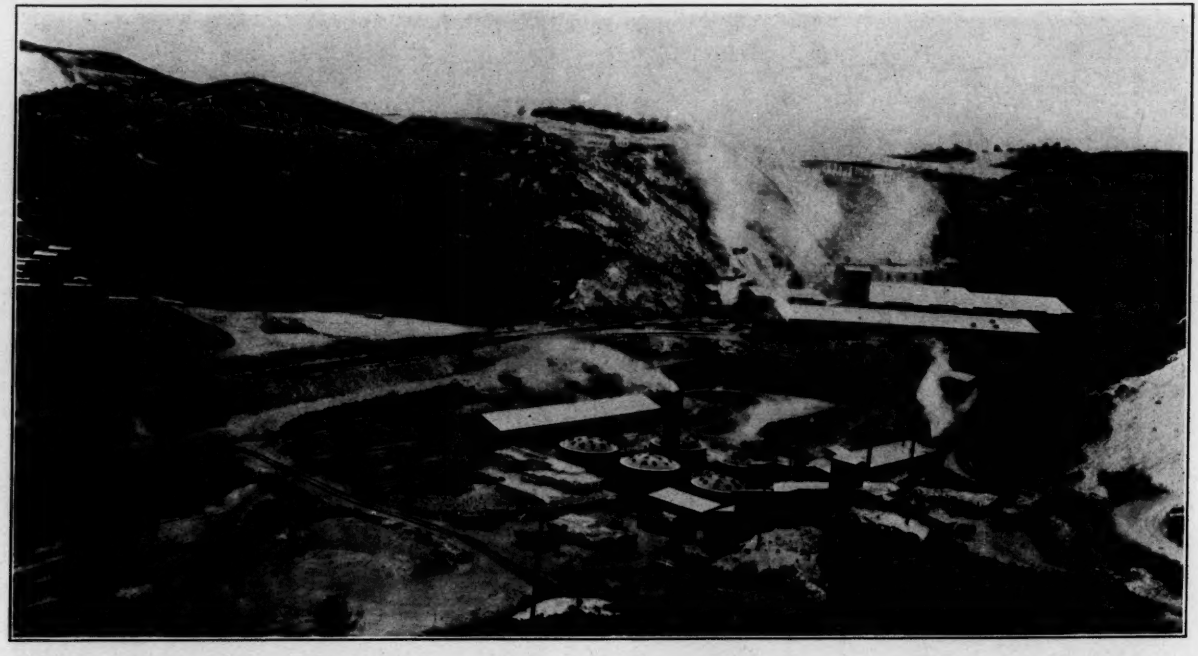


FIG. 3. —PORTION OF PLANT AND PROPERTY OF THE CELITE PRODUCTS COMPANY, WHITE HILLS (NEAR LOMPOC) CALIFORNIA. KILNS FOR BURNING C–22 BRICK IN FORE-GROUND.

Over 200 varieties of diatoms in this deposit have been classified. The fact that certain specific types of diatoms found in the Lompoc deposit do not occur elsewhere indicates that this particular deposit originated from a different source than the other known deposits. It is, in fact, a marine deposit (originating from diatoms living in the sea) and for purposes of identification the Lompoc material may be referred to as "Plankton Marine."

Laid down with the diatoms are found the remains of many large and small fishes as well as an extinct variety of duck. Portions of a mammoth and also of a whale have been found. Most of these fossil bones have been sent to Stanford University.

It is of interest to recall that one of the theories of the origin of petroleum is that it was formed as a result of oils which are secreted in diatoms. The diatom is the only plant which lays up in its basic tissue a supply of reserve food in the form of oil; as much as 10 to 50 per cent of its volume consisting of a peculiar oily substance from which petroleum is thought to be formed. The resulting accumulations of petroleum did not necessarily remain in the diatomaceous earth "beds," but probably drained off to lower levels through sand or other porous strata.

The material is found in many other places throughout the country, but not, as a rule, of such exceptional purity and freedom from clay as in the California deposit, nor in such large quantities with a sufficiently light overburden as to render its commercial exploitation readily practicable. It is of interest to note that certain deposits along the Patuxent River in Maryland are at the present writing indicating more or less active development.

The material in powdered form has many important applications other than for insulating purposes, such for example as a filtering material in sugar refineries and elsewhere, as an admixture for improving the workability of concrete, especially

for roads, as a light mineral filler for certain products such as automobile polish, silver polish, diluting talcum powder, etc., and as a nitro-glycerine absorbent in some grades of dynamite.

As a heat insulation the material has, in general, a somewhat lower insulating value than "85 per cent magnesia" in most of the forms in which it is produced commercially. However, in the natural state it resists any deteriorating affect of temperature up to about 1600 degrees F., and when calcined at a sufficiently high temperature gives very satisfactory insulating results up to a temperature slightly below the temperature at which it is calcined. The material may be calcined to about 2800 degrees F. without materially breaking up the structure of the diatoms; its melting point is about 2930 degrees F.

As a pipe covering for high temperature steam the natural diatomaceous earth in powdered form and bonded with long fiber asbestos, magnesium or calcium carbonates, etc., to form a sectional or segmental pipe covering, together with a similar plastic material, are both suitable for temperatures of 750 degrees F., which temperature is the highest to which tests of this material have been carried by the Navy.

A block material, similar in character to that from which the sectional pipe covering is made, is considered the most suitable material for the insulation of surfaces up to 1500 degrees F. This temperature classification includes all parts of the tube casings and uptakes of boilers which are not exposed to furnace temperatures. A block material of this character 2-1/2 inches thick, and laid up with a plastic of the same material so as to form, as nearly as practicable, a monolithic wall held in place by sheet steel lagging is required by the Bureau of Engineering on all new construction of oil-fired boilers.

A natural diatomaceous earth brick, that is, a standard brick 9 by 4-1/2 by 2-1/2 inches, composed of pure diatomaceous earth without asbestos or artificial bond of any kind,

is supplied commercially. The brick are cut directly from selected strata of the natural mineral. These brick have a very low thermal conductivity and are excellent for many purposes, but they are satisfactory only for temperatures up to 1600 degrees F., about 100 degrees F. more than the block and, being somewhat more difficult to handle due to a tendency to split along stratification planes, they are not approved for shipboard installations in the Navy as a tube casing insulation.

The calcined brick, also 9 by 4-1/2 by 2-1/2 inches, is manufactured by molding and pressing to shape a plastic of more or less coarsely powdered calcined diatomaceous earth, a suitable binder, and water, and burning same to the temperature desired in a special kiln. This brick being light in weight, semi-refractory, and with an insulating value several times that of ordinary refractory firebrick, makes the ideal insulation for the refractory lined furnaces of oil-fired boilers. The higher the calcination temperature the less the shrinkage, but also the less the insulating value. So it is advisable to keep the calcination temperature as low as practicable consistent with satisfactory service results. To reduce service shrinkage, therefore, the calcination temperature should be about 50 degrees F. above the highest temperature the insulation is expected to reach in service; at the present time the highest service temperature expected for the brick is about 2400 degrees F.

A further form of the calcined diatomaceous earth material is supplied commercially, viz., the coarse granular material. This material has been in use for some time as the insulation under the refractory lining of the furnace floors of oil-fired boilers. At present, however, this material is being displaced for this purpose by the calcined brick. This material when mixed with about 20 per cent of Portland cement to form an insulating concrete, has, however, given excellent service as a replacement for the refractory linings of cleaning doors of B. and W. and similar types of boilers; it is a good insula-

tion and about one-third the weight of the refractory lining which it replaces.

The calcined insulating brick are laid up in a single 4-1/2 inch course back of the usual 4-1/2 inch course of refractory firebrick, and a thin wash of a suitable diatomaceous earth cement is used for the joints to give a suitable monolithic wall of insulation. The side, front, and back wall firebrick are secured by bolting extending through the insulating brick to the outer steel casing; the insulating brick may be readily drilled for the bodies of the refractory brick bolts. It has been found advisable to drill the holes for the bolts in insulating brick and casing slightly oversize as an aid in replacement, and to place an asbestos millboard gasket under the nut on the outside of the casing as a further aid to insure integrity of the insulation.

The efficiency of a boiler depends on the differential temperature between the water in the boiler and the flue gases and the amount of radiant energy given off by the combustion of the fuel. Radiation from a boiler setting lowers the differential temperature and hence the boiler efficiency. Radiant energy is absorbed by the refractory brickwork and is lost by radiation from the settings unless measures are taken to prevent it. Insulation back of the refractory furnace linings reduces the heat loss and causes the refractory to act as a thermal reservoir resulting in better thermal control and higher combustion efficiency. But in causing the refractory to act as a thermal reservoir the temperature is materially increased throughout its thickness and therein lies the limitation in applying insulation to oil-fired boiler furnaces.

Owing to weight and space considerations, 4-1/2 inches of refractory is considered all that can reasonably be required in Naval boilers and such thickness has given satisfactory service. Refractory walls of this thickness, however, must be secured by bolting to withstand vibration, pulsation, and shock to which Naval boilers are subject. As refractory walls

are insulated the temperature increases materially and so the limit of the design is in the metal used for the brick bolts. Navy furnace temperatures are above the melting point of practically all metals. The refractory temperature is very close to the melting point of most available materials, and with insulation the temperature gradient through the refractory is much less steep. Regardless of the melting point most metals oxidize heavily at elevated temperatures and their life is short. Months of experiment have not solved the problem but the most satisfactory available bolt material to date is considered to be manganese-nickel, about 96 per cent nickel with 4 per cent of manganese. This material is readily produced by casting, and, owing to the small shank and large head of the brick bolts, this is considered the most satisfactory and economical method.

Attempts have been made to air cool the bolts by hollow shanks, radiating fins on the nuts, etc., but without success. Extensive air cooling of course defeats the purpose of the insulation and therefore cannot be considered. In any case the bolt head must be kept as far from the furnace face of the refractory as possible and the bolt head must be protected by the full refractory, that is, plugged holes in refractory over the bolt heads cannot be permitted.

Apart from diatomaceous earth materials, other heat resistant insulations are available commercially. Asbestos in one form or another is often supplied, but while it is heat resistant it is not in most of its commercial forms a good insulator; as a multi-ply or felted covering it is more or less satisfactory, but the life of the asbestos fiber decreases rapidly at temperatures at which the water of crystallization is lost; the water of crystallization is lost by the fiber at comparatively low temperatures but it is not until temperatures of 400 degrees F. or 500 degrees F. are reached that the loss has a serious affect in shortening the life of the insulation. In no case should asbestos be used at temperatures above 1000 degrees F. where any dependence must be placed on the strength of the fiber.

Certain special methods are used in bonding magnesium carbonate so that this material also is quite satisfactory as high temperature insulation within reasonable limits; such material is at present furnished by only one manufacturer.

Other interesting materials of recent development are the fibrous insulations made up from slag or rock fiber; a material made from a zinc slag fiber is of particular interest and has a limited distribution commercially. But these materials are not well standardized, are largely proprietary, and not strictly applicable to a large percentage of the service installations.

THEORY OF THE IMPELLER.

By Albert E. Guy, Associate Member, Mechanical and Hydraulic Engineer.

FUNDAMENTALS.

The present Theory of the Impeller is based upon the following assumptions and considerations:

1—For one given regime of operation the power applied to the impeller shaft is uniform and constant.

2—It follows that the torque, or turning moment, is constant.

3—It follows also that the angular velocity is constant.

4—The impeller shaft is horizontal, but no account is taken of the effect of gravity along the vertical diameter of the Impeller.

5—While in the generality of cases the flow of water through the pump, from the inlet to the outlet, is produced by the action of the Impeller, H_t being the total head required, and h_f the flow head, $(H_t - h_f)$ will be considered here as the head directly generated by the Impeller. In actual practice, of course, the Impeller will be designed for a head of H_t feet.

6—In conformity with experience, the flow into the Impeller proper, at the first element of the vanes, will be taken in a radial direction.

7—The flow into the Impeller proper takes place in concentric layers, each expanding gradually from the inlet periphery to the outlet periphery. Each layer, considered infinitely thin, will cover a cylindrical surface whose total area will be the sum of all the cylindrical segments included between the vanes and the side walls of the Impeller.

8—The total area of one given cylindrical layer multiplied by the radial velocity of flow obtaining for that layer will represent the volume of liquid traversing this cylindrical surface per unit of time, usually taken as the second.

Reciprocally, knowing the length of the arc of circle comprised between two adjacent vanes, the radial velocity of flow and the volume passing per second, the axial distance, or width of passage, between the side walls may be obtained by dividing the volume by the product of the arc by the radial velocity and by the number of vanes.

9—For any layer of radius R the velocity diagram will be as shown in Figure 1.

c is the radial velocity;

v is the circumferential velocity;

w is the relative velocity;

u is the absolute velocity;

ω is the actual velocity of entrainment of the liquid. All velocities are in feet per second.

10—On a circumference of radius R, a being the area of an infinitely small section of the layer, the volume flowing through is $a \times c$ per second. If the flow was assumed as taking place in the direction either of w or of u, the corresponding section of passage would be either a_w or a_u, each of these being measured normally to the respective direction of flow. The same volume passing per second it follows that $a \times c = a_w \times w = a_u \times u$.

11—The radial velocity may be kept uniform from the inlet to the outlet, or it may be varied to suit the ideas of the designer.

12—Whatever may be the radial velocity, its variation must be smooth. The passage from layer to layer must be without abruptness and the cross sections must succeed one another in such a way that the same volume per unit of time will traverse each of them without suffering any abrupt expansion or contraction, without interruption and without voids. This condition is of prime importance and insures the principle of continuity of flow.

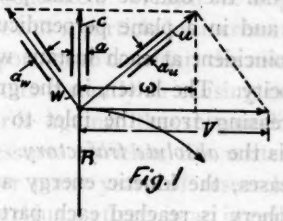


Fig. 1

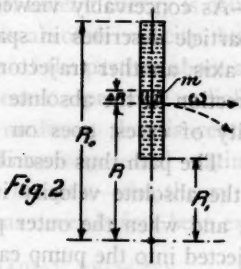


Fig. 2

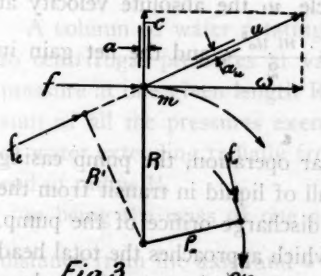


Fig. 3

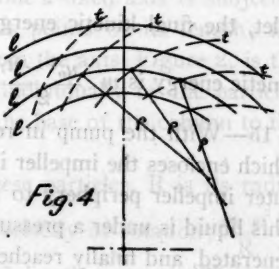


Fig. 4

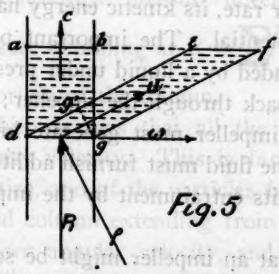


Fig. 5

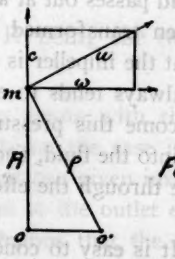


Fig. 6

13—Within the Impeller, and always in contact with the side and vane walls, the fluid may be considered as composed of an infinity of particles, each of which in its trajectory from the inlet to the outlet, follows one definite trajectory which has the same form as a vane, assuming for the sake of convenience that each vane is reduced to the thickness of a line. This is the *relative trajectory.*

14—As conceivably viewed from the outside of the pump, each particle describes in space, and in a plane perpendicular to the axis, another trajectory coincident at each instant with the direction of the absolute velocity. The latter, in the great majority of cases, goes on increasing from the inlet to the outlet. The path thus described is the *absolute trajectory.*

As the absolute velocity increases, the kinetic energy augments, and when the outer periphery is reached each particle is projected into the pump casing with an absolute velocity u_o; m being the mass of the particle, u_1 the absolute velocity at inlet, the final kinetic energy is $\frac{m\,u_o^2}{2}$, and the net gain in kinetic energy is $m\,\frac{u_o^2 - u_1^2}{2}$.

15—With the pump in regular operation, the pump casing which encloses the impeller is full of liquid in transit from the outer impeller periphery to the discharge orifice of the pump. This liquid is under a pressure which approaches the total head generated, and finally reaches it at the discharge orifice, where the liquid passes out at a regular rate, its kinetic energy having now been transformed into potential. The important points are: that the impeller is surrounded by a liquid under pressure which always tends to escape back through the impeller; that to overcome this pressure the impeller must generate kinetic energy into the fluid, and that the fluid must furnish additional pressure through the effects of its entrainment by the impeller vanes.

16—It is easy to conceive that an impeller might be so designed that, while rotating at a given rate of revolutions, the flow could take place without any hindrance from the vanes, and each particle would then follow a strictly radial trajectory. There would be no gain imparted to the fluid, either of pressure or of kinetic energy. But while keeping the flow steady the revolutions per minute might be increased; then a different condition would obtain: all along each vane rotary motion would be imparted to the successive layers, and energy would

be generated. Within the impeller, kinetic energy is communicated to a particle by increasing its absolute velocity, but most important to consider, the rotation of the particle about an axis results instantly in causing it to exert a pressure radially outwards against the surrounding medium; that is centrifugal pressure. Within the writer's knowledge—and stating this after many years of research on his part—the theory of the centrifugal effects in an impeller has never before been exposed.

PART I.—UNIFORM RADIAL VELOCITY.

A column of water rotating about a fixed axis is subjected to centrifugal pressures at various points of its length; the pressure at one given length R from the axis, Figure 2, is the sum of all the pressures exerted individually by the particles of water extending radially from the base of the column to the end of radius R.

m being the mass of one of these particles, R is its radial distance from the axis, and ω its velocity of rotation; $\frac{m\ \omega^2}{R}$ is the centrifugal pressure developed by that mass. The total pressure obtaining at R is $\sum_{R_1}^{R} \frac{m\ \omega^2}{R}$, R_1 being the radius at the origin of the column.

This assumes that all the particles rotate with the same angular velocity. This is far from being the case in actual practice, and of the particles forming at one given instant one radial column extending from the inlet to the outlet each has its own angular velocity, and it is obvious that the pressure equation must be established so as to take account of the true variation of velocity.

Let the pressure at end of radius R be $\sum_{R_1}^{R} p_c$, c being the centrifugal pressure developed by one given particle.

For a very small increase ΔR in radius, the pressure at the end of the radius $(R + \Delta R)$ will be $\sum_{R_1}^{R+\Delta R} p_c$. The difference

between these two values: $\sum_{R_1}^{R+\Delta R} p_c - \sum_{R_1}^{R} p_c$, is evidently $p_c = \frac{m\,\omega^2}{R}$.

If the particle was made to move from R to R + ΔR, it would have to overcome the difference in pressures obtaining at the two points, and the work required to make the move would be: $p_c \times \Delta R$.

MOMENTS ABOUT THE AXIS.

A particle, of mass m, at end of radius R, Figure 3, is acted upon by a force f, at right angles to R. The effect of this force is to increase the velocity of rotation ω possessed by the mass.

At radius R the centrifugal force developed by the particle m is $p_c = \frac{m\ \omega^2}{R}$.

The particle is caused to move radially outwards at the rate of c feet per second, which is the radial velocity of flow. But, since it travels simultaneously in a perpendicular direction (to R) at the rate of ω feet per second, the resultant travel is done with a velocity of u feet per second.

In a small amount of time dt the travel along the direction of u is udt.

But udt is the resultant of cdt and ωdt. In traveling along ω the work required is that concerning the change of velocity from ω to $\omega + d\omega$, the fluid pressure remaining constant along ωdt.

The travel along the radius takes place because the velocity c is assumed to be produced by causes outside of the system. Therefore the change in velocity does not involve inertia work. In the present case there is no change involved, c being considered as uniform radially. Later on, however, a varying radial flow will be considered, but in both cases the changes in radial velocity, or the velocity itself, will be as if produced by outside causes.

This question of velocity c applies to the mere radial travel. In changing, however, from R to R + dR the particle must exert a pressure p_o along dR. But since the particle is made to follow the absolute path along u, which is oblique to ω, the cross section of the absolute channel will now be a_u instead of that a of the radial path. Therefore the equivalent pressure f_o along u, of p_o along c will be: $f_o = p_o \frac{a_u}{a}$.

Obviously, because of similar triangles, $\frac{a_u}{a} = \frac{c}{u}$, whence $f_o = p_o \frac{c}{u}$.

The lever arm of f_o is R′.

Now, the motion is communicated to the system by a constant force f_o acting tangentially on a circumference of radius ρ_o. This force moves at a uniform rate, the angular velocity being $\varphi = \frac{\pi n}{30}$, where n = revolutions per minute. Equilibrium is expressed by the equation of moments: $f_o\rho_o = f_oR' + fR$.

ELEMENTARY WORK.

During the element of time dt the travels of the forces are: $\rho_o\varphi\, dt$, $\omega\, dt$, $u dt$, and the consequent work is expressed by the equation $f_o(\rho_o\varphi dt) = f\omega dt + f_o u dt$.

Since the final result must be to impart to the particle m a given energy, the total work may be represented by the product of the weight mg of the particle multiplied by the height to which it may be raised, or total work = mgH.

Since the prime moving force f_o is constant and its angular velocity φ is uniform, the work per unit of time will be uniform, therefore the elementary work in time dt will be

$$mgdh = f_o(\rho_o\varphi dt) = f\omega dt + f_o u dt.$$

The force f produces an increase $d\omega$ of velocity ω in time dt, then: $f\, dt = m\, d\omega$; whence $f = m\frac{d\omega}{dt}$.

Therefore $f\,\omega\,dt = m\,\frac{d\omega}{dt}.\quad \omega\,dt = m\,\omega\,d\omega.$

Because $f_c = p_c\,\frac{c}{u}$ and $p_c = \frac{m\,\omega^2}{R}$, therefore:

$$f_c\,u\,dt = \frac{m\,\omega^2}{R}.\ \frac{c}{u}.\quad u\,dt = \frac{m\,\omega^2}{R}\,c\,dt.$$

Let p = weight of the particle of mass m, then $mg = p$, and $p\,dh = mg\,dh = m\,\omega\,d\omega + \frac{m\,\omega^2}{R}\,c\,dt$, whence

$$g\,dh = \omega\,d\omega + \frac{\omega^2}{R}\,c\,dt. \qquad \text{Eq. 1.}$$

The problem is solved when the value of ω is found, since the data required for the delineation of the vane and of the trajectory are but functions of ω.

REMARK.

It has been stated that the flow through the impeller takes place in concentric layers. A most important point is that of the pressure obtaining on the surface of each layer. In Figure 4 all the layers l are traversed by the absolute trajectories t, which are all alike. Therefore on layer l_o, for instance, all the trajectories have the same curvature, and the instantaneous radius of curvature ρ is alike for all. It follows that each mass m following a trajectory must be under the same conditions of pressure and velocity when crossing the given layer l_o. Then, as previously stated, the layers may be considered as disposed concentrically, with a uniform pressure obtaining all over each cylindrical surface.

It may be pointed out that since the elementary mass m follows a curved trajectory, the centrifugal pressure developed at any point of the path might be considered a function of the radius of curvature obtaining at that point rather than of the actual radius of the layer containing the given point. The work equation will now be established accordingly and compared with that already obtained.

The cross section of the elementary mass m of liquid may be represented, when traveling radially, by the rectangle $a\ b\ g\ d$, Figure 5. In traveling in the direction of u, the surfaces dg and ef of the layer remaining parallel, the axial thickness being constant in both cases, it is obvious that the volume of the mass is not different from that in the first case. Thus:

$$\frac{\text{volume } a\ b\ g\ d}{\text{volume } g\ d\ e\ f} = \frac{dg \times ad}{g'g \times de} = 1, \text{ since } \frac{g\ g'}{dg} = \frac{c}{u}, \frac{ad}{de} = \frac{c}{u}.$$

However, the centrifugal pressure developed along R and which acted normally upon dg, will now be replaced by another developed along the instantaneous radius of curvature ρ. Since the fluid pressure, per unit of area, upon a fluid at a given point is the same in all directions, the total pressure p_t exerted normally on the cross section gg' of the trajectory canal is to the total pressure p_ρ upon dg, taken as a unit, in the ratio:

$$\frac{gg'}{dg} = \frac{c}{u}, \text{ or } \frac{p_t}{p_\rho} = \frac{c}{u}.$$

In Figure 6, a point at end of radius R rotates about center o with the velocity ω while at the same time sliding radially outward with the velocity c.

The resultant velocity u may be considered as produced by the rotation of m at the end of a radius ρ about an instantaneous center o′.

o′ is situated on a parallel to ω passing through o. The centrifugal pressure thus developed is $\frac{m u^2}{\rho} = p\rho$; ρ being perpendicular to u. Therefore

$$\frac{\rho}{R} = \frac{u}{\omega}, \quad \rho = R\frac{u}{\omega}.$$

$$p_\rho = \frac{m u^2}{\rho} = \frac{m u^2}{\frac{R u}{\omega}} = \frac{m \omega u}{R}.$$

$$\frac{p_t}{p_\rho} = \frac{p_t}{\frac{m \omega u}{R}} = \frac{c}{u}. \quad \text{Therefore } p_t = \frac{m \omega c}{R}.$$

The work of the centrifugal pressure during the time dt is

$$p_t \times u\,dt = \frac{m\,\omega\,c}{R}.\ u\,dt = \frac{m\,\omega\,u\,c\,dt}{R}.$$

The change of velocity from u to $u + d\,u$ requires the work $m\ u\ du$. Therefore the total work done during the instant dt is

$$m\left(\frac{\omega\,u\,c\,dt}{R} + u\,du\right) = p\,dh, \text{ where } p \text{ is the weight of } m.$$

Whence $$\frac{\omega\,u\,c\,dt}{R} + u\,du = g\,dh.$$

The difference between this value and that previously obtained in Equation 1, lies in the fact that the latter was established upon the assumption that the radial velocity c was produced by outside causes, whereas for the equation just obtained the velocity c is developed at each instant by the impeller itself. This requires that $\int (g\ dh)$ be somewhat greater than with the first method. It is considered preferable to adopt finally the assumption established in paragraph 5 of the Fundamentals throughout this work for two reasons, one of which being that in some cases the flow of water arriving at the pump inlet is produced by outside causes, while the other is that the task of handling the mathematical development of the first formula is much simpler. As stated before, in practical application the flow head will be added to that generated by the impeller, according to requirements.

DETERMINATION OF ω.

It was stated above that the problem is solved when the value of ω is found. While the importance of this point may be fully realized as the development of this subject proceeds, it is well to elucidate it now. The factor ω varies from zero at the beginning, to a maximum at the end of the vane; the tangent to the vane at any point within the impeller depends upon the value: of the radial velocity of flow c, of the circum-

ferential velocity v, and of ω, all obtaining at that point situated at the end of a radius R. Therefore, to delineate the vane it is necessary first to calculate a series of values for ω, corresponding to a series of radii extending from the inner to the outer circumference of the impeller; then, knowing the corresponding c and v, the angular direction of the vane for each ω is readily established, the corresponding radius of curvature of the vane is calculated, and with it an arc of vane is delineated between two adjacent points. A succession of such arcs gives the required form of the vane, after which the axial section of passage through the impeller is very easily calculated. According to this method there is continuity and smoothness of action upon each particle of water in such a manner as to secure the desired effect with a minimum of imput energy.

The condition of maximum efficiency requires that there should be no shock imparted to a single particle of water in its traject from the inlet edge of the vane to the outlet edge; consequently the water should enter without disturbance. Unfortunately, owing to the thickness at the inner edge, the radial direction at entrance is somewhat disturbed and the continuity of the flow interfered with. This question will be analyzed later on, but for the present, and in accordance with custom, it will be assumed that the direction of entrance is strictly radial.

At the inner circumference, of radius R_1, from which all the vanes begin, the circumferential velocity is v_1, Figure 7, the radial velocity is c_1, and the relative velocity, tangent to the first element of the vane, is w_1. Consequently $\omega_1 = o$, and c_1 equals the absolute velocity u_1. At the outer periphery the velocities are respectively: v_o, c_o, w_o, ω_o and u_o, all in feet per second.

Taking up the fundamental equation: $g\,dh = \omega\,d\omega + \frac{\omega^2}{R}\,c\,dt$, and making $c\,dt = dR$, dt may be removed from that expression. Thus: $g\,dh = \omega\,d\,\omega + \frac{\omega^2\,dR}{R}$. Eq. 2.

It may be noted here that $\frac{m\,\omega^2\,dR}{R}$ represents the centrifugal work performed along the distance dR.

But it is also possible to express dR as a function of H and of R_1 and R_o, the limit radii. Thus, the total head H is developed during the total radial travel $R_o - R_1$; since this travel is at an uniform rate c feet per second, and the increase in head is also as a direct function of the time, therefore

$$\frac{dR}{R_o - R_1} = \frac{dh}{H}, \text{ whence } dR = \frac{R_o - R_1}{H}.\,dh.$$

Integrating between the limit R_1 and a given R to which the height h corresponds:

$$\int_{R_1}^{R} dR = \left(\frac{R_o - R_1}{H}\right)\int_o^h dh = \left(\frac{R_o - R_1}{H}\right).\,h.$$

$$\therefore R - R_1 = \frac{R_o - R_1}{H}.\,h, \text{ and } R = \frac{R_o - R_1}{H}.\,h + R_1$$

Using this value of R in Equation 2:

$$\frac{dR}{R} = \frac{R_o - R_1}{H}\,\frac{dh}{\left(\frac{R_o - R_1}{H}\right)h + R_1} = \frac{dh}{\frac{(R_o - R_1)\,h + R_1 H}{R_o - R_1}}$$

$$= \frac{dh}{h + \frac{R_1 H}{R_o - R_1}}$$

For the present the constant $\left(\frac{R_1 H}{R_o - R_1}\right)$ may be replaced by κ, thus: $\frac{dR}{R} = \frac{dh}{h + \kappa}$.

The work equation: $g\,dh = \omega\,d\omega + \frac{\omega^2\,dR}{R}$ may now be written

$$g\,dh = \omega\,d\omega + \omega^2\frac{dh}{h + \kappa}.$$

This expression may be brought to the form of a linear equation of the first degree. Thus, remarking that $2\,\omega\,d\,\omega$ is the differential of ω^2, making $\omega^2 = y$ and $2\,\omega\,d\,\omega = d\,y$, we have:

$$2\,g\,dh = 2\,\omega\,d\omega + 2\,\omega^2 \frac{d\,h}{h + \kappa} = dy + 2\,y\,\frac{d\,h}{h + \kappa}.$$

Let

$$2g = \mathrm{Q} \text{ and } \frac{2}{h + \kappa} = \mathrm{P}.$$

Then

$$\mathrm{Q}\,dh = d\,y + \mathrm{P}\,y\,dh \qquad \text{(A)}$$

Assume $y = uz$, and let $dh = d\,x$.

$$\therefore \frac{d\,y}{d\,x} = \frac{u\,d\,z + z\,d\,u}{d\,x} = u\,\frac{d\,z}{d\,x} + z\,\frac{d\,u}{d\,x}.$$

Substituting in (A):

$$u\,\frac{d\,z}{d\,x} + z\,\frac{d\,u}{d\,x} + \mathrm{P}\,u\,z = \mathrm{Q}.$$

or

$$u\,\frac{d\,z}{d\,x} + \left(\frac{d\,u}{d\,x} + \mathrm{P}\,u\right) z = \mathrm{Q} \qquad \text{(B)}$$

Assume $\frac{d\,u}{d\,x} + \mathrm{P}\,u = o \quad \therefore \frac{d\,u}{u} = -\,\mathrm{P}\,d\,x$

Integrating:

$$\int \frac{d\,u}{u} = -\int \mathrm{P}\,d\,x$$

$$\log.\, u = -\int \mathrm{P}\,d\,x + c_1.$$

Whence

$$u = c_1\,\epsilon^{-\int \mathrm{P}\,dx} \qquad \text{(C)}$$

Then (B) becomes: $u\,\frac{d\,z}{d\,x} = \mathrm{Q}$

Replacing u by its value in (C):

$$c_1\,\epsilon^{-\int \mathrm{P}\,dx}\,\frac{d\,z}{d\,x} = \mathrm{Q}.$$

$$c_1\,d\,z = \mathrm{Q}\,\epsilon^{\int \mathrm{P}\,dx}\,d\,x + c.$$

Whence

$$z = \int \frac{\mathrm{Q}\,\epsilon^{\int \mathrm{P}\,dx}\,d\,x + c}{c_1}.$$

Therefore

$$y = uz = c_1 \epsilon^{-\int P dx} \int \frac{Q \epsilon^{\int P dx} dx + c}{c_1}$$

and finally:

$$y = \epsilon^{-\int P dx} \int Q \epsilon^{\int P dx} dx + c \qquad (D)$$

Replacing P and dx by their values:

$$\epsilon^{\int P dx} = \int \epsilon^{\frac{2}{h+K} dh} = \epsilon^{2 \log (h+K)}$$

The limits are $x = h$ and $x = o$.

Hence

$$\epsilon^{\int P dx} = \epsilon^{2 \log (h+K) - 2 \log K} = \epsilon^{2 \log \left(\frac{h+K}{K}\right)}$$

Then,

$$\int Q \epsilon^{\int P dx} dx = Q \int \epsilon^{\int P dx} = Q \int \epsilon^{2 \log \left(\frac{h+K}{K}\right)} dh$$

Let

$$m = \epsilon^{2 \log \left(\frac{h+K}{K}\right)} \quad \therefore m = \left(\frac{h+K}{K}\right)^2$$

$$\therefore \int \left(\frac{h+K}{K}\right)^2 dh = \frac{1}{K^2} \int \left[h^2 dh + 2 K h dh + K^2 dh\right]$$

Integrating between limits $h = h$ and $h = o$:

$$\int \left(\frac{h+K}{K}\right)^2 dh = \frac{1}{K^2} \left[\frac{h^3}{3} + K h^2 + K^2 h\right].$$

Let

$$\epsilon^{-\int P dx} = \epsilon^{-2 \log \left(\frac{h+K}{K}\right)} = \epsilon^{\log \left(\frac{K}{h+K}\right)^2} = n$$

$$\log n = \log \left(\frac{K}{h+K}\right)^2 \quad \therefore \epsilon^{-\int P dx} = \left(\frac{K}{h+K}\right)^2$$

Equation (D) may now be written:

$$y = \frac{K^2}{(h+K)^2} \cdot \frac{2g}{K^2} \left[\frac{h^3 + 3 K h^2 + 3 K^2 h}{3}\right]$$

By adding $\frac{K^3}{3}$ to the expression between brackets and subtracting $\frac{K^3}{3}$ from the total member on the right the value of the equation will not be altered, but we will then have:

$$\frac{h^3 + 3\,\kappa h^2 + 3\,\kappa^2 h + \kappa^3}{3} = \frac{(h+\kappa)^3}{3}$$

Hence:

$$y = \frac{2}{3} g \frac{(h+\kappa)^3 - \kappa^3}{(h+\kappa)^2} = \frac{2}{3} g \left((h+\kappa) - \frac{\kappa^3}{(h+\kappa)^2} \right);$$

and finally, since $y = \omega^2$:

$$\omega = \sqrt{y} = \sqrt{\frac{2}{3} g \left[(h+\kappa) - \frac{\kappa^3}{(h+\kappa)^2} \right]}.$$

Obviously:

$$\frac{h}{H} = \frac{R - R_1}{R_o - R_1} \quad \therefore h = \frac{R - R_1}{R_o - R_1} H.$$

But

$$\kappa = \frac{R_1 H}{R_o - R_1}, \text{ and } (h+\kappa) = \left(\frac{R - R_1}{R_o - R_1} + \frac{R_1}{R_o - R_1} \right) H$$

$$= \frac{R}{R_o - R_1} \cdot H$$

Combining these values:

$$\frac{\kappa^3}{(h+\kappa)^2} = \frac{\left(\frac{R_1 H}{R_o - R_1} \right)^3}{\left(\frac{R H}{R_o - R_1} \right)^2} = \frac{R_1^3}{R^2 (R_o - R_1)} \cdot H$$

Finally we obtain:

$$\omega = \sqrt{y} = \sqrt{\frac{2}{3} g H \left[\frac{R}{R_o - R_1} - \frac{R_1^3}{R^2 (R_o - R_1)} \right]}$$

$$= \sqrt{\frac{2}{3} g H \frac{R^3 - R_1^3}{R^2 (R_o - R_1)}}$$

For the maximum value of ω, $R = R_o$, consequently:

$$\omega_{max} = \Omega = \sqrt{\frac{2}{3} g H \frac{R_o^3 - R_1^3}{R_o^2 (R_o - R_1)}} \qquad \text{Eq. 3}$$

THE K_ω DIAGRAM

The formula just established may be employed as it is by the designer but the task of the latter can be made much easier by the use of a simple diagram such, for instance, as that given here on Chart 1.

Let H = total head, for radius R_o

h = head for radius R.

h varies as $R - R_1$, thus $\frac{h}{H} = \frac{R - R_1}{R_o - R_1}$

$$\therefore H = h \frac{R_o - R_1}{R - R_1}.$$

Introducing this value of H in Equation 3, gives:

$$\omega = \sqrt{\frac{2}{3} g h . \frac{R_o - R_1}{R - R_1} \left(\frac{R^3 - R_1^3}{R^2 (R_o - R_1)} \right)}$$

$$= \sqrt{\frac{2}{3} g h \frac{R^3 - R_1^3}{R^2 (R - R_1)}}. \qquad \text{Eq. 4.}$$

Let

$$\frac{R}{R_1} = n, \text{ then } \frac{R^3 - R_1^3}{R^2 (R - R_1)} = \frac{n^3 - 1}{n^2 (n - 1)}.$$

The value of g usually adopted is 32.18 feet.

The formula becomes:

$$\omega = K_\omega \sqrt{h}, \text{ where } K_\omega = \sqrt{\frac{2 \times 32.18}{3} \cdot \frac{n^3 - 1}{n^2 (n - 1)}}.$$

n	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8
K_ω	7.6607	7.36405	7.11689	6.90816	6.72981	6.57585	6.44175	6.32402
n	1.9	2.	2.1	2.2	2.3	2.4	2.5	2.6
K_ω	6.21991	6.12726	6.04431	5.9697	5.90223	5.84095	5.78508	5.73395
n	2.7	2.8	2.9	3.	3.1	3.2	3.3	3.4
K_ω	5.68699	5.64372	5.60274	5.5667	5.53228	5.50023	5.47031	5.44233
n	3.5	3.6	3.7	3.8	3.9	4.		
K_ω	5.4161	5.39141	5.3683	5.34646	5.3256	5.30639		

These figures were used in the delineation of the diagram. The latter gives the values of K_ω plotted as ordinates against the ratios $\frac{R}{R_1} = n$ as abscissas.

In practice the radial space is divided into 10 parts, this makes it easy to establish the corresponding radii. Thus, with $(R_o - R_1) = 10$ units, the successive values of $(R - R_1)$ will be: 1, 2, 3, 4, 5, 6, 7, 8, 9, 10 units; and the corresponding values of h will be very simply:

$$\frac{H}{10}, \frac{2H}{10}, \frac{3H}{10}, \text{ etc.}$$

For example, assume $R_o = 10$, $R_1 = 5$, $H = 100$.

$$n = \frac{10}{5} = 2 \quad K_\omega = 6.127 \quad \Omega = 6.127 \sqrt{100} = 61.27$$

for an intermediate radius

$$R = 8, \frac{R}{R_1} = \frac{8}{5} = 1.6 = n, \quad K_\omega = 6.576$$

$$h = \frac{6}{10}, \ 100 = 60, \ \omega = 6.576\sqrt{60} = 50.936.$$

Later on a vane will be computed completely in a practical manner so as to illustrate clearly the use of the diagram and the handling of the other factors connected with the problem of impeller design.

PART II.—VARYING RADIAL VELOCITY.

The reasons for employing a radial velocity varying from the inlet to the outlet of the impeller are twofold. In the first place, long before the centrifugal pump became an efficient apparatus, investigators sought, constantly and in many ways, to improve it, one of which was to so design the impeller passages that the water would leave the periphery with a low velocity, just equal to that obtaining in the delivery part of the pump casing. It was thought that the shock losses resulting from the change of kinetic energy possessed by the leaving water into potential energy would be, if not totally eliminated, at least enormously reduced. An axial section of the impeller would then be somewhat as represented in the sketch, Figure 8.

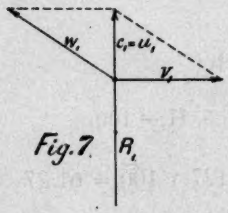


Fig. 7.

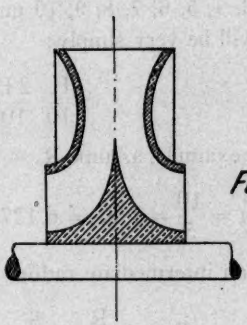


Fig. 8.

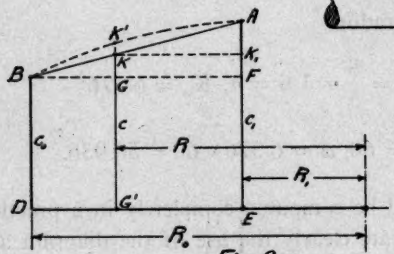


Fig. 9.

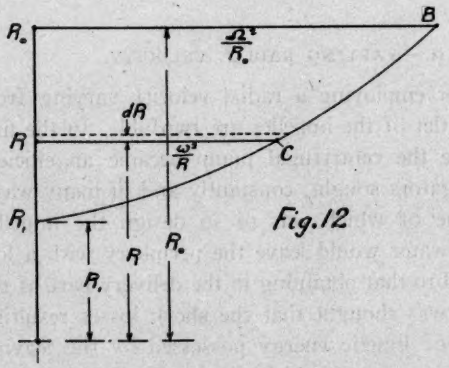


Fig. 12.

This scheme was tried at various times and may be considered now as abandoned, although some twelve years ago it was still thought of, as evidenced by the exaggerated patent claims which were then prominently circulated. The whole work of the impeller consisted, it was claimed, in generating only centrifugal pressure. This assumed, of course, that a method has been established for calculating this pressure with neat precision, so as to permit the delineation of the proper form of blading. To the writer's knowledge such a method has never been published.

The second reason is of a practical order and founded on something really substantial. The one drawback to the more general use of centrifugal pumps was originally the difficulty of securing high speed prime movers. Of course, to quite an extent, high speed could be obtained through belt and pulley transmission, but even to date, this is used only when other means are lacking; the inherent difficulties are well known to engineers.

The advent of the steam turbine and the development of the electric motor actually brought the centrifugal pump to life and made it really the useful apparatus so well appreciated in many industrial fields. At the beginning, however, many of the apparatus were of small size, handling small quantities of water. It was quickly realized that the axial width of the water passages through the impeller, when designed for a constant radial velocity, would be so small ($\frac{1}{4}$ inch, $\frac{1}{8}$ inch at the periphery in some cases) that it would be materially impossible to make the sand cores required for foundry work. For quite some time impellers were made with a central web on which the vanes were attached, and to which side plates were riveted. Trouble occurred, however, through breaking of the rivets on account of vibration, or corrosion of the rivets when acidulated water was handled. Then again, the side joints were not always perfect, and leakage would occur from one vane space to another. All this led to the use of the varying radial velocity by means of which ampler widths of passage were secured.

A curious thing happened in the hydraulic tests of some pumps. It was found that in many cases the performance curve would drop more or less abruptly after passing the point of maximum efficiency, and to suit which the impeller had been specifically designed. This fact proved to be useful, in that it prevented automatically the overloading of the prime mover. One concern advertised widely its exclusive ability to so design its apparatus as to insure the entire safety of the prime mover throughout the range of the pump delivery. The curious point, however, was that its own people did not know exactly how it was done, and could not logically explain the phenomenon; with a certain suitable ratio of $\frac{c_1}{c_o}$ being adhered to in every case, the advertised effect would invariably take place. The question will receive due attention in another section.

At the origin E, Figure 9 (the first element of the vane), the radial velocity is c_1, and at the end D (periphery of the impeller) it is c_o. It is assumed that the difference $(c_1 - c_o)$ decreases uniformly from E to D. This decrease may be arranged in several ways to suit the designer's ideas. For many years the writer simply joined the points A and B by a straight line, so that the velocity for a point at end of radius R would be:

$$c = c_o + (c_1 - c_o)\frac{R_o - R}{R_o - R_1}.$$

With the present method another form has been adopted which is more in accord with the general principle upon which this theory is based. The difference is assumed to decrease uniformly from E to D *directly as the times.* Therefore, if DE is taken as the scale of times, the length DE will represent the total time of traverse T; EG′ will be any chosen time t. The two similar triangles give $\frac{AK_1}{KK_1} = \frac{AF}{BF}$, or $\frac{c_1 - c}{E} = \frac{c_1 - c_o}{T}$. Line AB locus of the end of ordinate c is bound to be straight.

But if the values of c are plotted for different *spaces* traversed from E to D, which spaces are measured on the scale of the radii R_1 to R_o, and do not vary directly as the times, then the locus of the c ordinates will be on a curve concave towards the R scale, as indicated by the dotted line.

To fix the ideas, take for example: $R_o = 12$ inches, $R_1 = 6$ inches, $c_1 = 10$ feet, $c_o = 7$ feet. The total time of traverse

$$= \frac{R_o - R_1}{\left(\frac{c_1 + c_o}{2}\right)} = T.$$

$$t = \frac{R - R_1}{\left(\frac{c_1 + c}{2}\right)} \quad \therefore c = 2\frac{(R - R_1)}{t} - c_1$$

$R_1 = 6$ inches $= \frac{1}{2}$ foot. $R_o = 1$ foot. $T = \frac{1 - 0.5}{\left(\frac{10 + 7}{2}\right)} = \frac{1}{17}$ second.

Divide T into five equal periods: $\frac{1}{85}, \frac{2}{85}, \frac{3}{85}, \frac{4}{85}, \frac{5}{85}$, seconds. Correspondingly $(R_o - R_1)$ will be divided into five unequal spaces.

$$\frac{c_1 - c}{t} = \frac{c_1 - c_o}{T} \quad \therefore c = c_1 - (c_1 - c_o)\frac{t}{T}.$$

Let

$$t = \frac{n}{85} \quad \therefore \frac{t}{T} = \frac{n}{5}$$

With these data we have:

$t =$	0	$\frac{1}{5}$	$\frac{2}{5}$	$\frac{3}{5}$	$\frac{4}{5}$	$\frac{5}{5}$	
$c =$	10	9.4	8.8	8.2	7.6	7.	feet
$\frac{c_1+c}{2} =$	10	9.7	9.4	9.1	8.8	8.5	feet
$t =$	0	$\frac{1}{85}$	$\frac{2}{85}$	$\frac{3}{85}$	$\frac{4}{85}$	$\frac{5}{85}$	seconds
$\frac{c_1+c}{2} \times 12t =$	0	1.37	2.655	3.855	4.97	6	inches
$\frac{c_1+c}{2} \times 12t + R_1 =$	6	7.37	8.655	9.855	10.97	12	=R in inches

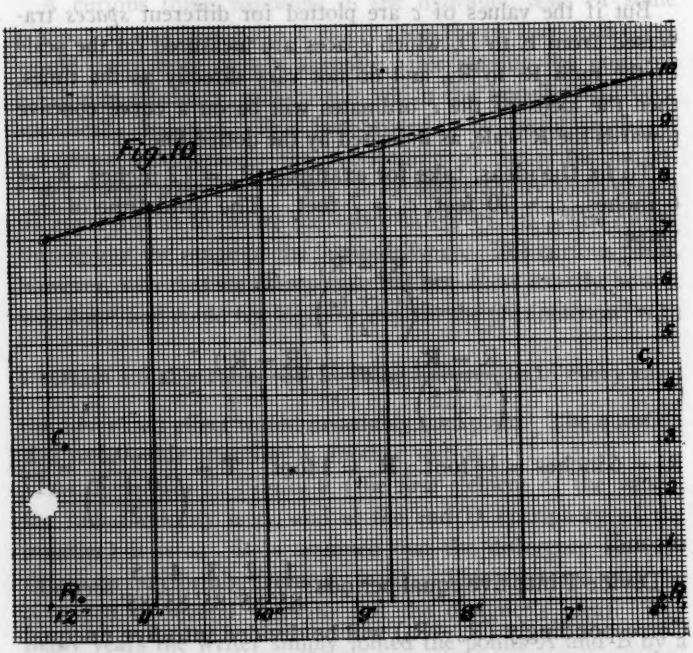


Fig. 10
C_2
C_1
R_2
12"
11"
10"
9"
8"
7"
6"

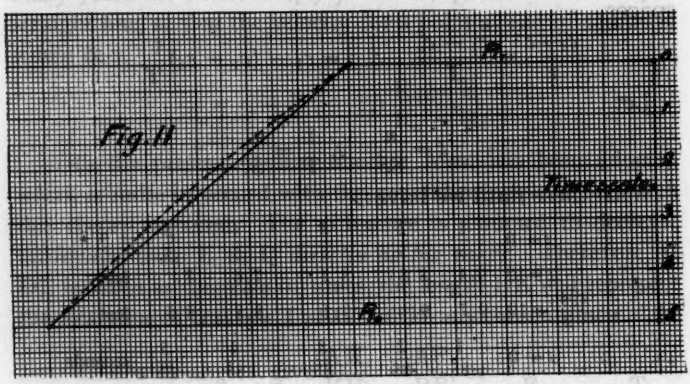


Fig. 11
R_2
Time scale
R_1

With these figures the diagrams, Figures 10 and 11, are drawn, showing respectively the curve of c on the R scale, and the curve of R on the time scale. These curves do not deviate very far from the straight lines connecting the extremities.

Having adopted this method, we have: $c = c_1 - (c_1 - c_o)\frac{t}{T}$, or

$$c = \frac{c_1 T - (c_1 - c_o)\, t}{T}.$$

Any radius $R = \left(\frac{c_1 + c}{2}\right) t + R_1$, whence, with the above value of c:

$$R = \frac{2\,T\,c_1 t - (c_1 - c_o)\, t^2 + 2\,R_1 T}{2\,T}$$

The work equation found for the first case (constant radial velocity) applies also to this one, that is: $g\,dh = \omega\,d\omega + \omega^2 \frac{c\,dt}{R}$. But c and R may be expressed in function of t, and $\frac{c\,dt}{R}$ preferably transformed in function of h, so as to keep only two variables in the equation.

Now, h varies directly as t; this may be expressed thus:

$$\frac{t}{h} = \frac{T}{H} \quad \therefore t = \frac{T}{H} \cdot h \text{. whence, also, } dt = \frac{T}{H}\,dh.$$

Replacing t by its value in h in the c and R equations, we have:

$$c = \frac{c_1 T - (c_1 - c_o)\, t}{T} = \frac{c_1 H - (c_1 - c_o)\, h}{H} \qquad \text{Eq. 5}$$

$$R = \frac{2\,c_1 T\left(\frac{T}{H} h\right) - (c_1 - c_o)\left(\frac{T}{H} h\right)^2 + 2\,R_1 T}{2\,T}$$

$$= \frac{2\,c_1 T H h - (c_1 - c_o)\, T h^2 + 2\,R_1 H^2}{2\,H^2}$$

Consequently:

$$\frac{c\,dt}{R} = \frac{c_1 H - (c_1 - c_o)\,h}{H} \cdot \frac{\frac{T}{H}\,dh}{\frac{2\,c_1 T H h - (c_1 - c_o)\,T h^2 + 2\,R_1 H^2}{2\,H^2}}$$

or

$$\frac{c\,dt}{R} = \frac{[c_1 T H - (c_1 - c_o)\,T h]\,2\,dh}{2\,c_1 T H h - (c_1 - c_o)\,T h^2 + 2\,R_1 H^2}$$

Making, as before, $\omega^2 = y$, $2\,\omega\,d\omega = dy$, then:

$$2\,g\,dh = 2\,\omega\,d\omega + \omega^2\,\frac{2\,c\,dt}{R} = dy + y\,\frac{2\,c\,dt}{R}.$$

$\frac{2\,c\,dt}{R}$ may be put in a more convenient form, thus:

$$\frac{2\,c\,dt}{R} = \frac{\left[2\left(\frac{2c_1 H}{c_1 - c_o}\right) - 4\,h\right]dh}{\frac{2\,c_1 H}{c_1 - c_o}\,h - h^2 + \frac{2\,R_1 H^2}{(c_1 - c_o)\,T}}$$

Now, let $a = \frac{2\,c_1 H}{c_1 - c_o}$, $b = \frac{2\,R_1 H^2}{(c_1 - c_o)\,T}$.

We have finally:

$$2\,g\,dh = dy + y\left[\frac{(4\,h - 2\,a)\,dh}{h^2 - a\,h - b}\right].$$

Following the same method as in Part I, again make

$$2\,g = Q, \text{ and } \frac{4\,h - 2\,a}{h^2 - a\,h - b} = P.$$

Then $\frac{dy}{dh} + P\,y = Q$.

Solving for y gives:

$$y = \epsilon^{-\int P\,dh} \int Q\,\epsilon^{\int P\,dh}\,dh + C$$

The two integrations:

$$\left(-\int P\,dh\right) \text{ and } \left(\int P\,dh\right)$$

must be solved first.

$$\int P\,dh = \int \frac{4h-2a}{h^2-ah-b}\cdot dh = \int\left[\frac{4h\,dh}{h^2-ah-b} - \frac{2a\,dh}{h^2-ah-b}\right].$$

Factoring the denominator: $h^2-ah-b=o$

$$h=\frac{a}{2}\pm\sqrt{b+\frac{a^2}{4}},\ h'=\frac{a}{2}+\frac{\sqrt{4b+a^2}}{2},\ h''=\frac{a}{2}-\frac{\sqrt{4b+a^2}}{2}$$

$$\therefore h^2-ah-b=(h-h')(h-h'')$$

$$=\left(h-\frac{a+\sqrt{4b+a^2}}{2}\right)\left(h-\frac{a-\sqrt{4b+a^2}}{2}\right).$$

Let

$$\frac{4h-2a}{h^2-ah-b}=\frac{A}{h-\frac{a+\sqrt{4b+a^2}}{2}}+\frac{B}{h-\frac{a-\sqrt{4b+a^2}}{2}}$$

Developing and solving, we have A = B = 2, whence

$$\int P\,dh=\int\frac{4h-2a}{h^2-ah-b}\,dh=\int\frac{2\,dh}{h-\frac{a+\sqrt{4b+a^2}}{2}}$$

$$+\int\frac{2\,dh}{h-\frac{a-\sqrt{4b+a^2}}{2}}$$

$$\int_{h=o}^{h=h} P\,dh=\log.\left(\frac{h^2-ah-b}{b}\right)^2$$

$$\epsilon^{-\int P\,dh}=\left(\frac{b}{h^2-ah-b}\right)^2 \qquad \epsilon^{\int P\,dh}=\left(\frac{h^2-ah-b}{b}\right)^2$$

$$\int Q\,\epsilon^{\int P\,dh}\,dh=\frac{Q}{b^2}\int(h^2-ah-b)^2\,dh$$

and

$$y=\frac{2g}{(h^2-ah-b)^2}\int(h^2-ah-b)^2\,dh$$

Integrating again between the limits $h = h$ and $h = o$, then developing and arranging, we have finally:

$$y = 2g.\frac{\frac{h^5}{5} - \frac{2ah^4}{4} + \frac{a^2 - 2b}{3}h^3 + 2a\frac{bh^2}{2} + b^2h}{(h^2 - ah - b)^2} = \omega^2 \quad \text{Eq. 6}$$

The constants a and b must now be dealt with.

$$a = \frac{2c_1H}{c_1 - c_o}; \quad b = \frac{2R_1H^2}{(c_1 - c_o)T}.$$

T may be replaced by its value: $T = \frac{2(R_o - R_1)}{c_1 + c_o}$, and then

$$b = \frac{R_1(c_1 + c_o)H^2}{(c_1 - c_o)(R_o - R_1)}.$$

With these values introduced in Eq. 6, and solving for $\omega_{max} = \Omega$ corresponding to $h = H$, we have:

$$\Omega^2 = 2gH\left[\frac{\frac{8c^2_1+9c_1c_o+3c_o^2}{15(c_1+c_o)^2} - \frac{(4c_1^2+6c_1c_o+2c_o^2)}{3(c_1+c_o)^2}\left(\frac{R_1}{R_o-R_1}\right)+\left(\frac{R_1}{R_o-R_1}\right)^2}{\left(1+\frac{R_1}{R_o-R_1}\right)^2}\right]$$

The numerator may be changed into a perfect square plus a fractional number, thus:

$$\Omega^2 = 2gH\left[\frac{\left[\frac{R_1}{R_o-R_1}+\frac{1}{3}\left(1+\frac{c_1}{c_1+c_o}\right)\right]^2+\frac{1}{45}\left(4-\frac{c_1c_o}{(c_1+c_o)^2}\right)}{\left(1+\frac{R_1}{R_o-R_1}\right)^2}\right]$$

It may be easily shown that for $c_1 = c_o$, that is, for uniform radial velocity, the formula becomes:

$$\Omega = \sqrt{\frac{2}{3}gH\frac{R_o^3 - R_1^3}{R_o^2(R_o - R_1)}},$$

which is Equation 3 found previously.

For regular practice it is advisable to select a uniform ratio $\frac{c_1}{c_o}$ and adhere to it for all applications. For many years the writer used the ratio $\frac{c_1}{c_o} = \frac{10}{7}$, because the resulting performance curve gave a satisfactory control of the Horse Power input for the range past the point of maximum efficiency. On certain occasions, however, some other ratio was used, better suited to meet special requirements. The work of the designer will be minimized by the use of prepared diagrams such as Charts 2, 3, 4, 5, 6, made to cover a range of values of the coefficient K_ω corresponding to a range of values of the ratio $\frac{R_o}{R_1} = n$, extending from 1 to 4. Thus for a given value of $\left(\frac{R_o}{R_1} - 1\right)$ a coefficient K_ω is read off the diagram, and then $\Omega = K_\omega \sqrt{H}$.

We thus have the same kind of formula as in Part I, with K_ω made up of the factors under the radical. For $\frac{c_1}{c_o} = \frac{10}{7}$

$$K_\omega = \sqrt{\frac{2g \cdot \left[\frac{R_1}{R_o - R_1} + \frac{1}{3}\left(1 + \frac{10}{10+7}\right)\right]^2 + \frac{1}{45}\left(4 - \frac{10 \times 7}{(10+7)^2}\right)}{\left(1 + \frac{R_1}{R_o - R_1}\right)^2}}$$

In the following table values of K_ω are given for the ratios

$$\frac{c_1}{c_o} = \frac{10}{10}, \frac{10}{9}, \frac{10}{8}, \frac{10}{7}, \frac{10}{6}, \frac{10}{5}.$$

With these the practician may very readily construct his own diagrams, preferably on much larger scale than that of those given here. Very rarely the ratio $\frac{R}{R_1}$ extends to 4.

The values of K_ω in the Table were calculated, with the exception of those for the ratio $\frac{10}{10}$, from the following formulas:

$$\frac{c_1}{c_o} = \frac{10}{9} \quad K_\omega = \sqrt{\frac{2g\left[\frac{1}{n-1} + 0.50877\right]^2 + 0.083347}{\left(1 + \frac{1}{n-1}\right)^2}}$$

$$\frac{c_1}{c_o} = \frac{10}{8} \quad K_\omega = \sqrt{\frac{2g\left[\frac{1}{n-1} + 0.51852\right]^2 + 0.083402}{\left(1 + \frac{1}{n-1}\right)^2}}$$

$$\frac{c_1}{c_o} = \frac{10}{7} \quad K_\omega = \sqrt{\frac{2g\left[\frac{1}{n-1} + 0.52941\right]^2 + 0.0835}{\left(1 + \frac{1}{n-1}\right)^2}}$$

$$\frac{c_1}{c_o} = \frac{10}{6} \quad K_\omega = \sqrt{\frac{2g\left[\frac{1}{n-1} + 0.541667\right]^2 + 0.08368}{\left(1 + \frac{1}{n-1}\right)^2}}$$

$$\frac{c_1}{c_o} = \frac{10}{5} \quad K_\omega = \sqrt{\frac{2g\left[\frac{1}{n-1} + 0.55555\right]^2 + 0.08395}{\left(1 + \frac{1}{n-1}\right)^2}}$$

$$n = \frac{R_o}{R_1} \quad \text{or} \frac{R}{R_1}$$

$$g = 32.18$$

It should be understood, of course, that c_1 and c_o seldom occur in such reduced form as 10 and 7, for instance. The first determining factor is the velocity at the pump inlet, next it is the velocity at the impeller inlet proper, at the first element of the vane, or the c_1, dealt with thus far in this study. Assuming a case where $c_1 = 12.632$ feet per second; with a ratio of $\frac{10}{7} c_o = 12.632 \times \frac{7}{10} = 8.8424$. Obviously it is preferable to establish the ratio at once and then deal only with simple numbers in the calculation.

K_ω

$n = \frac{R}{R_1}$	$\frac{c_1}{c_0} = \frac{10}{10}$	$\frac{10}{9}$	$\frac{10}{8}$	$\frac{10}{7}$	$\frac{10}{6}$	$\frac{10}{5}$
1.1	7.6807	7.66709	7.6742	7.68221	7.69107	7.7012
1.2	7.36405	7.37577	7.38924	7.40334	7.41972	7.4385
1.3	7.11689	7.13296	7.15109	7.17121	7.1939	7.21904
1.4	6.90816	6.92817	6.95197	6.97535	7.0036	7.03516
1.5	6.72981	6.75312	6.77904	6.80802	6.84096	6.87774
1.6	6.57585	6.60202	6.63113	6.66363	6.70038	6.7420
1.7	6.44175	6.47046	6.50233	6.53801	6.57823	6.62416
1.8	6.32402	6.35491	6.38925	6.42778	6.47099	6.52014
1.9	6.21991	6.25273	6.28927	6.33013	6.37623	6.42814
2.	6.12726	6.16182	6.20031	6.24336	6.29191	6.34702
2.1	6.04431	6.08046	6.11221	6.16573	6.21644	6.27401
2.2	5.9697	6.00712	6.04897	6.09586	6.14855	6.20847
2.3	5.90223	5.94104	5.98427	6.03261	6.08712	6.14912
2.4	5.84095	5.87748	5.92198	5.97502	6.02827	6.09158
2.5	5.78508	5.82608	5.87172	5.92295	5.98066	6.04634
2.6	5.73395	5.77592	5.8227	5.87492	5.93412	6.00125
2.7	5.68699	5.72985	5.7771	5.83106	5.89142	5.95994
2.8	5.64372	5.68734	5.73608	5.79069	5.85205	5.92193
2.9	5.60374	5.64870	5.69767	5.75312	5.81568	5.88687
3.	5.5667	5.61180	5.6621	5.7184	5.78199	5.85408
3.1	5.53228	5.57809	5.62905	5.6861	5.75067	5.82398
3.2	5.50023	5.54658	5.59826	5.6561	5.72149	5.79578
3.3	5.47031	5.51830	5.56954	5.6281	5.69426	5.76948
3.4	5.44233	5.48974	5.54262	5.60185	5.66881	5.74484
3.5	5.4161	5.46402	5.51745	5.5767	5.64493	5.7218
3.6	5.39141	5.43984	5.49374	5.5542	5.6225	5.70013
3.7	5.3683	5.41710	5.47152	5.5325	5.60140	5.67975
3.8	5.34646	5.39566	5.45058	5.5121	5.58152	5.66052
3.9	5.3256	5.37542	5.43072	5.4927	5.56273	5.64237
4.	5.30639	5.35629	5.4120	5.4744	5.545	5.62521

PART III.—PROOF OF THE FORMULAS.

It is proposed here to calculate the elements of two impellers, one intended for a uniform radial velocity of 10 feet per second, that is, with $\frac{c_1}{c_0} = 1$; and another for a varying radial velocity with a ratio $\frac{c_1}{c_0} = \frac{10}{7}$. The inlet and outlet diameters are selected at random and equal 10 and 24 inches respectively. The total head produced at the point of maximum efficiency is taken as 100 feet in both cases. For the present no consideration is given to the number of revolutions per minute, assuming that it is constant and amply sufficient to insure, for any radius, a circumferential velocity greater than the corresponding ω. The question of R.P.M. as well as that of the volume handled by the impeller will receive special attention later on. This manner of proceeding is the reverse of that employed in treatises on pumps, in which the operating data is specified from the beginning. The advantage of the present method is that it enables one to determine correctly the working capacity of any impeller, merely by means of the dimensions of the latter.

FIRST IMPELLER.

Data:

$R_0 = 12$ inches. $c_1 = c_0 = 10$ feet per second.

$R_1 = 5$ inches.

Total head $H = 100$ feet; R.P.M. = constant.

PROBLEM: TO DETERMINE THE KINETIC AND THE CENTRIFUGAL PRESSURES OBTAINING AT ANY GIVEN POINT ALONG THE RADIAL PATH OF THE WATER.

Let the radial path $R_0 - R_1$ be divided into ten equal parts, the corresponding radii and the ratio of each radius to the inlet radius, or $\frac{R}{R_1}$, will be readily obtained. With these and

by means of the K_ω diagram the corresponding values of ω will be easily determined.

The total head h obtaining for each radius goes on increasing uniformly from 0 at R_1 to 100 at R_0.

The following table contains all the elements of the solution of the problem:

h_k is the kinetic pressure.

h_c is the centrifugal pressure.

The term $\left(\frac{\omega^2}{R}\right)$ will be used, as explained further, for the graphical integration of the values of the centrifugal pressures.

R	C	$n=\frac{R}{R_1}$	h	ω	$\frac{\omega^2}{R}$	h_k	h_c
5	10	1.	0	0.	0.	0.	0.
5.7	9.7	1.14	10	23.829	99.617	8.8225	1.1775
6.4	9.4	1.28	20	32.0336	160.337	15.94805	4.056
7.1	9.1	1.42	30	37.631	199.441	22.0017	7.9983
7.8	8.8	1.56	40	41.9624	225.749	27.3592	12.6408
8.5	8.5	1.70	50	45.561	244.095	32.2376	17.7624
9.2	8.2	1.84	60	48.6514	257.27	36.777	23.223
9.9	7.9	1.98	70	51.412	266 994	41.0696	28.9304
10.6	7.6	2.12	80	53.923	274.311	45.1786	34.8214
11.3	7.3	2.26	90	56.242	279.926	49.1481	40.8519
12.	7.	2.40	100	58.4095	284.306	53.009	46.991

Considering now the fundamental equation: $g\,dh = \omega\,d\omega + \frac{\omega^2\,dR}{R}$, it is clear that for one given R the total head generated will be:

$$h = \frac{\omega^2}{2g} + \int_{R=R_1}^{R=R_0} \frac{\omega^2\, dR}{g\, R}.$$

$\frac{\omega^2}{2g}$ is the kinetic pressure; $\int \frac{\omega^2 dR}{g R}$ is the centrifugal pressure, that is, the sum of all the increments of Centrifugal pressure generated from R_1 to R. This integration may be easily performed. From Equation 3:

$\omega^2 = \frac{2}{3} g H \frac{R^3 - R_1^3}{R^2 (R_o - R_1)}$, the value of $\frac{\omega^2}{g}$ may be written:

$$\frac{\omega^2}{g} = \frac{2}{3} H . \frac{1}{R_o - R_1} . \frac{R^3 - R_1^3}{R^2} .$$

Consequently

$$\int \frac{\omega^2 dR}{g R} = \left(\frac{2}{3} \frac{H}{R_o - R_1}\right) \int \frac{R^3 - R_1^3}{R^3} dR.$$

$\frac{R^3 - R_1^3}{R^3} dR = dR - \frac{R_1^3}{R^3} dR$, which integrated between the limits R_1 and R gives:

$$R - R_1 - \frac{R_1}{2} + \frac{R_1^3}{2 R^2}.$$

Finally:

$$\int_{R_1}^{R} \frac{\omega^2}{g} \frac{dR}{R} = \frac{H}{3} . \frac{2 R^3 - 3 R^2 R_1 + R_1^3}{R^2 (R_o - R_1)} = h_c =$$

centrifugal pressure for a given radius R.
For the same radius the kinetic pressure is

$$\frac{\omega^2}{2g} = \frac{H}{3} \frac{R^3 - R_1^3}{R^2 (R_o - R_1)} = h_k.$$

Of course the total pressure is $H = h_k + h_c$. The proof of these expressions is readily obtained, for instance, solving for $R = R_o$:

$$H = \frac{H}{3} \left[\frac{R_o^3 - R_1^3}{R_o^2 (R_o - R_1)} + \frac{2 R_o^3 - 3 R_o^2 R_1 + R_1^3}{R_o^2 (R_o - R_1)} \right]$$

After due cancellation this gives H = H; the problem stated above is therefore completely solved.

In the preceding table are inscribed the values of h_k and h_c derived from the above equations.

GRAPHICAL INTEGRATION.

Of greater interest perhaps is the method indicated below, by means of which the range in capacity of an impeller, whether designed according to this theory, or by any other may be easily analyzed. In practice it is at times possible to use an impeller for conditions of speed and head different from those for which it was originally made, by simply reducing its outer diameter, or for that matter, by increasing the latter through skillful foundry work. That is when the knowledge of the range is of greatest importance.

Having determined by actual measurements or by calculation the values of ω corresponding to a satisfactory number of radii R, a drawing may be made, as represented in Figure 12; the radii being marked off in inches, as ordinates, and the $\frac{\omega^2}{R}$ plotted correspondingly as abscissae, with a convenient scale, for instance 40 feet to the inch. The extremities of the abscissae should then be connected by a carefully drawn curve B C R_1. Then, with a planimeter, the area R C R_1 may be measured and read off in square inches. This should be done for each succeeding R; the final area measured being that of the segment R B R_1, corresponding to the total centrifugal head. Let A be one of these areas, and s the scale of abscissae, then $\frac{A \times s}{g} = h_c$ = centrifugal head for the radius R considered.

The simplicity of this result may be explained as follows. Let R C R_1 be a triangle, and reduce all dimensions to inches; then $\frac{(\omega \times 12)^2}{R''} \times \frac{R'' - R_1''}{2}$ = true area of triangle, in square inches.

$$\text{Area actually measured} = \frac{\text{true area}}{144 \times \text{scale}}.$$

$$\therefore \frac{\text{area measured} \times 144 \times \text{scale}}{g \times 12 \times 12} = h_c \text{ in feet} = \frac{As}{g}.$$

It is obvious that the diagram may be established directly with the values of $\frac{\omega^2}{g\,R}$, instead of $\frac{\omega^2}{R}$ as above.

The term $\frac{\omega^2\,dR}{g\,R}$ represents of course an area, but at the limit, dR may be considered so small that the area is reduced to the thickness of a straight line marked off as R C on the diagram; the sum of all the elementary areas comprised between the contour R C R_1 is in fact the sum of all the elementary centrifugal pressures obtaining between R_1 and R, and therefore is the actual centrifugal head for the radius R.

REMARK I.

For a new design of impeller the total head h_t assigned to each R being established from the start it is obviously unnecessary to calculate directly the centrifugal head. The latter may be obtained very simply from the expression $h_t = h_k + h_c$, whence

$$h_c = h_t - h_k = h_t - \frac{\omega^2}{2\,g}.$$

The preceding demonstrations have purposely been extended in details so as to firmly establish the correctness of the fundamental formulas, and it will be found that the practical applications require but a small amount of careful computation without the drudgery of mathematical work. This will be particularly obvious in the case of impellers with varying radial velocity.

REMARK II.

In the case of an old impeller to be analyzed it is necessary first to make a template of the contour of the forward face of the vane, and then to measure the areas of a number of sections of passage through the impeller between two adjacent vanes. From the template it will be easy to determine the vane angle corresponding to each of these sections. Assuming the radial velocity at the vane inlet to be 10 feet per second, the ratio of each sectional area to that at the inlet will give the radial velocity for each point considered.

At the inlet, and for delivery at point of maximum efficiency, the absolute velocity is radial [entrance of water without shock] and equal to c_1. Therefore by constructing the velocity diagram the circumferential velocity v_1 will be obtained. The v for any radius is equal to $\frac{V_1 R}{R_1}$. Hence the ω may be derived from a suitable number of diagrams, very easily drawn, so as to cover properly the whole range of the vane.

SECOND IMPELLER.

Data.

$R_o = 12$ inches. $c_1 = 10$ feet.

$R_1 = 5$ inches. $c_o = 7$ feet.

Total Head H = 100 feet; R.P.M. = constant.

Let the head generated increase uniformly, directly as the time of radial traverse, and let each increment equal $\frac{1}{10}$ of the total head, or $\frac{H}{10} = 10$ feet.

The values of the radial velocity and of the radii must be calculated for 9 intermediate points between R_1 and R_o.

From Equation 5 $\quad c = c_1 - (c_1 - c_o)\frac{h}{H}$.

$\frac{h}{H}$ = successively $\frac{1}{10}, \frac{2}{10}, \frac{3}{10} \ldots\ldots \frac{10}{10}$.

$$\therefore c = 10 - (10 - 7)\frac{h}{H} = 10 - \frac{3h}{H}.$$

$$R = \frac{c_1 + c}{2} t + R_1$$

But $\frac{t}{T} = \frac{h}{H}$ and $T = \frac{R_o - R_1}{\left(\frac{c_1 + c_o}{2}\right)}$.

$$\therefore t = T\frac{h}{H} = \frac{2(R_o - R_1)}{c_1 + c_o} \cdot \frac{h}{H}.$$

$$\therefore R = \frac{c_1 + c}{2} \cdot \frac{2(R_o - R_1)}{c_1 + c_o} \frac{h}{H} + R_1$$

$$R = R_1 + \left[2c_1\frac{h}{H} - (c_1 - c_o)\left(\frac{h}{H}\right)^2\right]\frac{R_o - R_1}{c_1 + c_o}. \quad \text{Eq. 7.}$$

$$R = 5 + \left[20\frac{h}{H} - 3\frac{h^2}{H^2}\right] \cdot \frac{7}{17}$$

It is next in order to calculate the ratio $\left(\frac{R_1}{R_o - R_1}\right)$ which is to be used in the ω equation.

From Equation 7 and dividing both members by R_1 we have:

$$\frac{R - R_1}{R_1} = \left[2c_1\frac{h}{H} - (c_1 - c_o)\frac{h^2}{H^2}\right]\frac{1}{c_1 + c_o} \cdot \frac{R_o - R_1}{R_1}.$$

$\frac{R - R_1}{R_1}$ may be put in the form $\left(\frac{R}{R_1} - 1\right)$, and by making $\frac{R}{R_1} = n$, $\frac{R - R_1}{R_1} = (n - 1)$, and $\frac{R_1}{R_o - R_1} = \frac{1}{n - 1}$.

Corresponding to each R the factor $\frac{1}{n - 1}$ is calculated and this value is introduced into the K_ω equation estab-

lished for the selected ratio $\frac{c_1}{c_0}$. Each of the K_ω thus obtained is multiplied by the square root of the corresponding h, and the result is the complete series of the values of ω. Thus $\omega = K_\omega \sqrt{h}$.

These operations entail quite a deal of work, which can be avoided by the use of such a diagram as that established for the ratio $\frac{c_1}{c_0} = \frac{10}{7}$.

In the present case it suffices to read on the diagram the value of K_ω corresponding to any factor $\left(\frac{1}{n-1}\right)$.

The following tabulation contains the result of the calculations discussed thus far, and in addition the $\frac{\omega^2}{R}$, h_k and h_e, as for the first impeller. The h_e were obtained by graphical integration because the formula for numerical

R	$\frac{R}{R_1} - 1$	c	$\frac{c}{c_1}$	ω	$\frac{\omega^2}{R}$	h	h_K	h_e
5.	0.	10	1.	0.	0.	0	0.	0.
5.8113	.16226	9.7	.97	23.726	96.872	10	8.7465	1.2535
6.5981	.31962	9.4	.94	31.887	154.1	20	15.7983	4.2017
7.3611	.47222	9.1	.91	37.53	191.345	30	21.8847	8.1153
8.09649	.61929	8.8	.88	41.989	217.775	40	27.294	12.706
8.8115	.7623	8.5	.85	45.728	237.315	50	32.4899	17.5101
9.4968	.89936	8.2	.82	49.032	253.155	60	37.3545	22.6455
10.159	1.0318	7.9	.79	52.0316	266.485	70	42.0648	27.9352
10.7988	1.15976	7.6	.76	54.7656	277.47	80	46.6014	33.3986
11.41116	1.2822	7.3	.73	57.281	287.54	90	50.9806	39.0194
12	1.4	7.	.70	59.75	297.505	100	55.4702	44.5298

integration is too difficult to handle for any practical purpose, and no attempt is made here to establish it, in view of the detailed demonstration given before of the correctness of the expression $h_t = h_k + h_e$.

PART IV.—DELINEATION OF THE VANE.

FIRST METHOD.

Considering now the velocity diagram Figure 13 for any radius R:

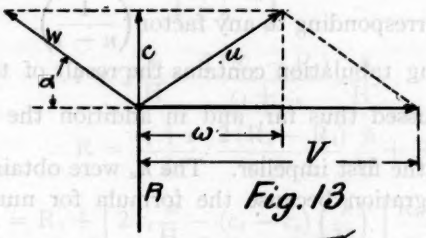


Fig. 13

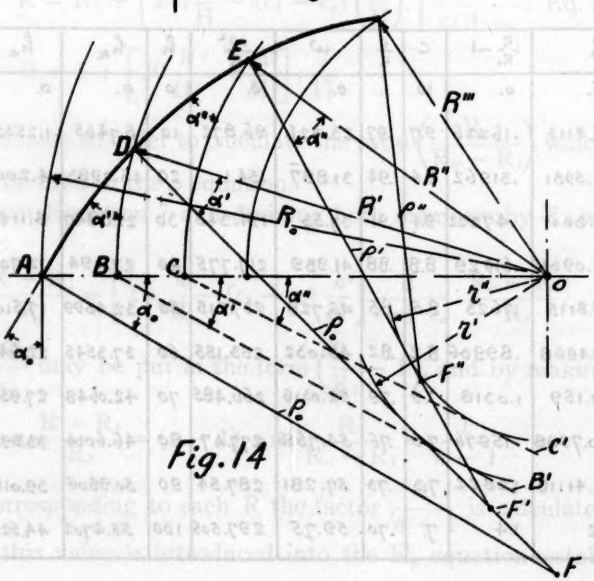


Fig. 14

c = the radial velocity of flow;
V = the circumferential velocity;
w = the relative velocity, along the vane;
u = the absolute velocity;
ω = the orthogonal projection of u upon V; it may be called the velocity of circumferential entrainment of the water.
α = angle of vane for radius R;
α_1 = angle of vane for radius R_1 at inlet;
α_o = angle of vane for radius R_o at outlet.

The problem for the first method consists in determining accurately the succession of values of α from the inlet to the outlet, and by means of these a series of radii of curvature for the elements of the forward curve of the vane.

The speed of rotation has, from the start, been considered constant. Let us assume that R.P.M. = 857. This number was obtained in the usual way by deciding upon a shut off pressure of 125 feet; this point will be dealt with at length subsequently.

$$V = 2\pi \frac{R}{12} \cdot \frac{R.P.M.}{60} = \text{feet per second.}$$

A formula much used is $V = \frac{DN}{229.18}$, where $D = 2R$, in inches, and N = R.P.M. This form is very convenient for slide rule computation.

V is therefore calculated for each radius.

Now, $(V - \omega) \tan \alpha = c$.

$$\therefore \tan \alpha = \frac{c}{V - \omega}, \text{ whence } \alpha.$$

Let O A, Figure 14, = R_o, B O = R′, C O = R″, etc.

A D, D E, etc., are successive elementary arcs; therefore A E is part of the contour of the vane. This curve forms with the circles R_o, R′, R″, the angles α_o, α', α'' . . .

Let ρ_o = the radius of curvature of A D, ρ' that of B E. Join O F and O D.

Consider the two triangles O F A, O F D; they have two sides respectively equal, and we may write:

$$\overline{O F}^2 = R_o^2 + \rho_o^2 - 2\rho_o R_o \cos \alpha_o = R'^2 + \rho_o^2 - 2\rho_o R' \cos \alpha'.$$

Whence $R_o^2 - R'^2 = 2\rho_o [R_o \cos \alpha_o - R' \cos \alpha']$

And finally:

$$\rho_o = \frac{R_o^2 - R'^2}{2 [R_o \cos \alpha_o - R' \cos \alpha']}.$$

The next radius of curvature, that for the arc D E is

$$\rho' = \frac{R'^2 - R''^2}{2 [R' \cos \alpha' - R'' \cos \alpha'']}$$

The vane angle α and the radius of curvature ρ are calculated for each of the radii R, determined as explained in the preceding sections.

It is very useful to draw the angles OAF, OBB′, OCC′, etc., because the origin of the first radius ρ_o is on AF, that of ρ' is at F′ on a radius ρ_o tangent to a circle which is tangent to BB′, side of angle α'; that F″ of radius ρ'' is on ρ' tangent to a circle tangent to CC′, side of angle α''; etc. Thus by drawing the auxiliary arcs of radii r', r'', etc., and being careful in insuring the tangency of the radii of curvature a most reliable control of the delineation process is insured. Each ρ must be correct in length; its origin must be on the preceding ρ; its extremity at the intersection of the preceding arc of vane with the corresponding concentric circle of radius R; and finally, it must be tangent to the concentric arc of radius r.

In spite of the apparent complexity of the process the delineation is easily performed and becomes routine work in practice.

SECOND METHOD.

This is based upon exactly the same principle as the first, only instead of computing the radii of curvature by means of the formula they are obtained by the cut and try process. This is of course more expeditious, and the skillful draftsman may obtain entirely satisfactory results.

The concentric circles of radii R_o, R′, R″, etc., are first drawn; then the corresponding angles a_o, a′, a″, etc., are properly located; finally the arcs of circles of tangency r′, r″, etc., are drawn. All of this preparatory work must be done with great care.

In Figure 15, the outermost vane element AD is shown. Suppose that the first radius tried be F′D′, having its origin on side AF of angle a_o; a straight line drawn lightly from D′,

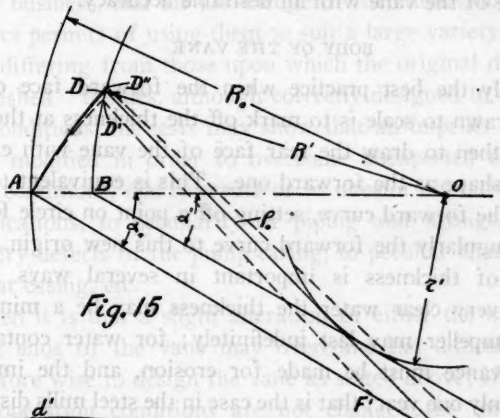


Fig. 15

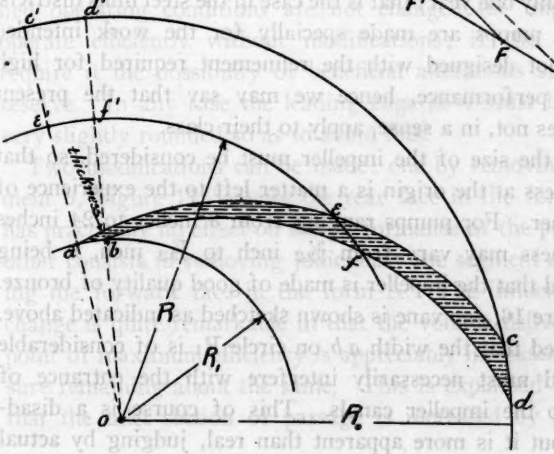


Fig. 16

at the intersection of arcs AD′ and BD′, to F′ should be tangent to arc r'. If it lies outside of the arc the radius F′D′ is too short. Another radius F″D″ being tried, line F″D″ is seen to pass inside the tangency arc r', therefore the second radius tried is too long. The correct radius ρ_0 originates at F, its arc intersects arc BD at D, and line FD is tangent to arc r'.

The process repeated for each of the vane elements will permit in a few minutes of time to draw the contour of the forward face of the vane with all desirable accuracy.

BODY OF THE VANE.

Probably the best practice when the forward face of the vane is drawn to scale is to mark off the thickness at the inlet end, and then to draw the rear face of the vane with exactly the same shape as the forward one. This is equivalent to first drawing the forward curve, setting off a point on circle R_1 and moving angularly the forward curve to this new origin. The question of thickness is important in several ways. For handling very clear water the thickness may be a minimum and the impeller may last indefinitely; for water containing grit, allowance must be made for erosion, and the impeller may last only one year; that is the case in the steel mills districts. Dredging pumps are made specially for the work intended and are not designed with the refinement required for high efficiency performance, hence we may say that the present theory does not, in a sense, apply to their class.

Again, the size of the impeller must be considered, so that the thickness at the origin is a matter left to the experience of the designer. For pumps ranging from 3 inches to 24 inches the thickness may vary from $\frac{3}{32}$ inch to $\frac{7}{32}$ inch, it being understood that the impeller is made of good quality of bronze.

In Figure 16, one vane is shown sketched as indicated above. It is noticed that the width $a\,b$ on circle R_1 is of considerable extent and must necessarily interfere with the entrance of water into the impeller canals. This of course is a disadvantage, but it is more apparent than real, judging by actual

results from hundreds of pumps. By extending *oa* to *c'* and *ob* to *d'* it is seen that for any R the circular thickness of the vane is $ef = e'f'$. This comes of use in the determination of the circular cross sections of passages throughout the impeller. The outer width *cd* appears also very large and likely to interfere seriously with continuity of the peripheral delivery.

Except in special cases an impeller is not made to suit solely one particular pumping proposition. Impellers are real assets in the business, and the knowledge of their individual characteristics permits of using them to suit a large variety of conditions differing from those upon which the original design was established. Besides, although correctly designed to suit specified conditions, the tests may show that an impeller may have to be modified in order to overcome unexpected changes in these conditions, due to errors or to ulterior alterations of the specifications, to peculiarity of piping and fittings, to some foundry defects of the pump casing, to peculiar characteristics of that casing, etc.

Then it is that a slight alteration of either the inlet or the outlet ends of the vane may overcome the difficulty. It is therefore wise to design the vane as stated above; if the specified operating conditions are not changed the impeller will operate efficiently without modifications; if the conditions require it the possibility of beneficial alterations stands as a reserve. In any case the leading edge at *a* must be smooth, very slightly rounded so as to avoid fins.

Two modifications can be made: one by removing the segment *b*, Figure 17, leaving the rear face in the form B; this has practically no effect on the performance of the pump. The other consists in removing judiciously the segment *a* and leaving the forward face in the form A. The influence of this change is quite remarkable in that the volume delivered at the point of maximum efficiency is appreciably increased, the pressure remaining about the same. This is explained by the fact that the inlet section of passage is increased by an amount

roughly estimated at $ab \times$ number of vanes $\times$ width of passage. Comparing conditions for the same volume Q delivered, it is seen from the diagram, Figure 18, that the radial velocity c_1 is slightly reduced; the vane angle α_1 has increased to α_1' and the water now enters with some shock with an absolute velocity u_1 not radial as originally designed.

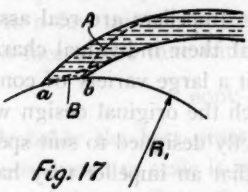


Fig. 17

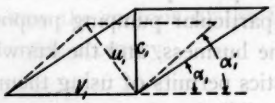


Fig. 18

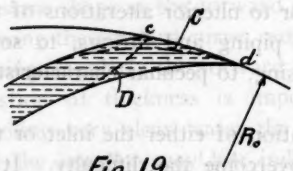


Fig. 19

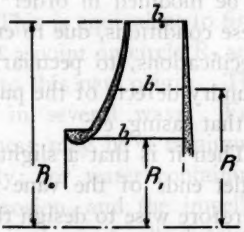


Fig. 20

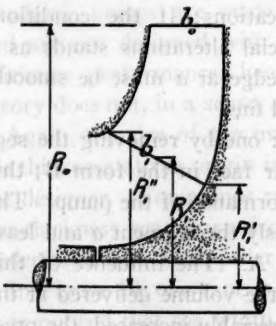


Fig. 21

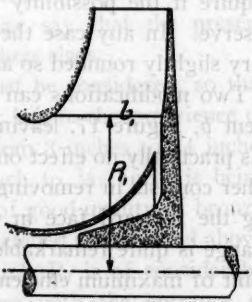


Fig. 22

At the outer end the vane may also be modified in two ways, either by removing the forward segment *c*, Figure 19, leaving the forward face in the form C, or by cutting out the segment *d*, making the rear face in the form D. The first change produces a reduction of the head generated, both through an increased discharge area, and a consequently reduced radial outlet velocity. The second produces an increase of head, mainly because the new shape increases the vane angle by a considerable amount.

In case of necessity both ends of the vane may be altered, but that must be done judiciously and be left to the experience of the designer, otherwise there is the risk of utterly spoiling a serviceable impeller and likewise it should be kept in mind that such changes as above indicated have a strong influence on the characteristics of the wheel.

AXIAL WIDTH OF VANE—NO. OF VANES.

For regular pump manufacture it is highly desirable to adopt a standard size for the inner radius of each size pump; this proves invaluable in estimating, in calculations and tabulations. The width *b*, Figure 20, varies from the inlet to the outlet as a function of R and of *c*; both these factors have been previously determined in this text.

The number of vanes is not such an important factor as mentioned by some authors. So far as the theory is concerned one vane only is sufficient. But the impeller must be a solid structure and the two sides must be well bound to one another. When the vane subtends a very wide angle from the origin to the outer end the number may be reduced to a minimum of four or five. For "shallow" wheels, that is, when the distance $R_o - R_i$ is relatively short, the number may be 10 or 12, so as to insure well defined canals. Generally speaking the number may extend from 5 to 10. A little experience and observation of good types will prove good guides in this matter. For pumps handling small quantities of water with an impeller of

relatively large outer diameter, difficulty may be experienced in the foundry because of too thin and brittle cores. Then a larger number of vanes may be so selected as to secure a desirable size for b_o, that is, for the thinnest part of the core.

If Q is the quantity of water, in gallons per minute, handled by the pump, then $\frac{Q \times 231}{60}$ = cubic inches per second.

The area of passage for a circumference of radius R is: $[2\pi R - n t_v]$ b = square inches, where $n t_v$ = circumferential thickness of vane × number of vanes.

The radial velocity is $c \times 12$ = inches per second. Therefore,

$$\frac{Q \times 231}{60} = [2\pi R - n t_v] \times b \times c \times 12,$$

whence

$$b = \frac{Q \times 231}{720\, c\, [2\pi R - n t_v]} = \frac{0.32084\, Q}{c\, [2\pi R - n t_v]}.$$

This formula holds for single suction impellers and also for the double suction type where each side handles the same quantity of water $= \frac{Q}{2}$; in the latter case the width on each side of the vertical axis is $\frac{b}{2}$. Special attention is required with this latter type to the proper shape of the central web so as to insure smoothness of flow. This web is rarely carried to the outer diameter of the wheel; it usually is somewhere midway between R_i and R_o. The terminal edge must be as thin as convenient and very smooth. It is desirable to make the web diameter a standard for each size of pump.

ENTRANCE INTO THE IMPELLER.

In many makes of pumps the impeller hub is so shaped as to form the long side of a directing canal, Figure 21, and the inlet edge of the vane is inclined normally to the general direc-

tion of the incoming current. These features are not desirable. The rotation of the hub produces a twist in the direction of flow of the body of water in contact with it and the entrance is not radial, as originally assumed; the point of maximum efficiency, which is always considered as a sort of guiding post in the design of a pump, is displaced one way or another, and this efficiency itself is somewhat impaired. It is not possible to analyze the true conditions of flow into the impeller so as to delineate the stream lines, because of the many factors involved. However, that the rotating hub surface has a detrimental effect upon the flow may be observed in one extreme case, that of the handling of hot water. Some fifteen years ago the writer made several boiler feed pumps for handling water at a temperature of about 230 degrees F. The head on the suction intake was about 7 feet. In this particular instance the available speed was very high, practically past the limit for the impeller size. The pumps tested in the usual way with cold water met the specifications in every respect and were then shipped to the customer. When operating under the intended conditions the delivery of each pump could not be raised over 800 gallons per minute, whereas at the test 1200 gallons per minute was readily obtained at the maximum efficiency point. During the investigation that followed it was found that, with a suitably made search tube introduced in the inlet passage, steam issued from the outer end of the tube when the inner end was located within a short distance from the hub surface. Proceeding then carefully further, the existence of an annular space filled with steam was clearly determined. It seems that the rotation of the hub must have, through the entrainment of the incoming mass of fluid, a sort of churning, as it were, caused the separation of vapor from the water, so that the intake was partially steam bound. As nearly as could be measured the steam area amounted to practically one-third of that of the intake, which readily accounted for the 400 gallons per minute shortage in delivery. To remedy this con-

dition the hub was shortened and turned down considerably; then, a fixed thin guiding cone, of the type shown in Figure 22, was made, having the same contour as the original hub, Figure 21, and with the hot water entering now without disturbance the capacity of the pump met the specifications very satisfactorily. Furthermore, it was found that the head upon the suction could be reduced to one-half, that is to about 3½ feet.

These experiments were so conclusive that all pumps were thereafter provided with suitable guide cones, and the additional cost was more than amply repaid by increased efficiency and regularity of performance. This leads to the general and most important statement that anything interfering with the entrance and passage of water into the suction main, into the pump casing, and finally into the impeller proper, may have a most serious effect upon the performance of the pumping unit. It may be added as a hint that a noisy pump is most likely to have defects in the suction end. It is then obvious that a correctly designed impeller may prove a failure if the entrance conditions are defective.

It is not necessary to have the guiding cone extended very far into the impeller inlet, as the water leaving the cone will continue in the direction of the last element of the latter, and then reach the inner edge of the vane without hindrance. This permits of an easier construction of the hub, Figure 22, of a consequent saving in material, and finally of the standardization of important parts.

It is well to analyze a little the position of the inner edge of the vane sketched in Figure 21; this vane was designed for radial action between R_1 and R_o. Here there are actually three R_1, viz: $R_1' < R_1$, R_1, and $R_1'' > R_1$ with the result: $(R_o - R_1') > (R_o - R_1) > (R_o - R_1'')$. The work done through $R_o - R_1'$ is much greater than that done through $R_o - R_1''$. Consequently a readjustment of pressure conditions within a short distance from the average entrance spot is bound to occur, which is bound to be detrimental to the work of the pump as a whole.

With impellers having a large $\frac{R_o}{R_1}$ ratio the disturbance is not so apparent, but again, in extreme cases, that is with shallow wheels, trouble of various kind is usually experienced. Some of it is attenuated through the introduction of suitable corrective coefficients in the formulas, but even that does not always prove entirely satisfactory.

The writer recollects in particular the case of one shallow impeller pump which was submitted to him some 14 years ago. It was a 12-inch pump, with a small $\frac{R_o}{R_1}$ ratio, high speed, usual form of rotating guiding hub surface, and a most pronounced angle of vane inlet edge. The $\frac{R_o}{R_1}$ ratio was obviously most undesirable. The pump was very much below its rated capacity and the customer had practically decided to reject the unit. The pump was then tested anew, and a search tube being introduced at various points of the incoming stream, close to the inlet edge, it was found this time that there were actually two currents of fluid in the inlet, as sketched in Figure 23; one in close contact with the hub, entering the wheel, and the other, along the outer webb surface, returning from the discharge chamber, passing through the wheel and re-entering the wheel inlet. Such a condition was deemed impossible by all concerned. Then another search tube was introduced in the discharge chamber, close to the periphery of the impeller. This tube was moved gradually across the leaving edge of the vanes, and it was found that two currents existed there also, one being the discharge from the wheel proper, extending from the midaxis towards the outer web, and the other returning from the discharge chamber, re-entering the wheel at the periphery and gliding against the inner surface of the outer web. The matter was then very clear, amounting to the case of two pumps working against one another, the more powerful one pushing its discharge clear through the impeller of the other.

The trouble was remedied by placing fixed shroud disks, one on each side of the impeller, and so constructed as to permit a jet action of the peripheral discharge, thus making full use of the kinetic energy of the water.

It is therefore evident that there is no gain in so shaping the inlet edge. The proper way is to make it parallel with the shaft axis. In the case cited above both the speed and the pump casing were unsuited for the specified conditions, and the so-called economy in size and original cost aimed at proved in the end a very expensive matter and a poor piece of business.

PART V.—DELIVERY END.

As soon as the water issues from the periphery of the impeller the work of the latter is of course accomplished. It has generated the required pressure and it should be evident, beyond question, that no part of the pump, no form of passage receiving the water can increase that pressure. However, quite a number of authors have held different opinions. Some hold that the discharge must be received in a volute surrounding the impeller periphery; others claim that there should be a diffuser passage; others again claim that there should be diffuser vanes in that passage, and that without either diffuser and diffuser vanes there is a tremendous loss of kinetic energy, and consequently that high pressure pumps are impossible without these appendages. Some have gone as far as to state that the diffuser, or whirlpool chamber, actually produced an increase of head, because of the supposed high speed rotation, within the diffuser passage, of the body of water discharged from the impeller, which of course was expected to produce additional centrifugal pressure.

Leaving aside the fact of the experience of the writer with some 3000 pumps, the question may be viewed from the common sense standpoint. Thus, power is applied to the shaft (S.H.P.), transmitted to the impeller, thence to the water, and the output (water H.P.) is measured directly past the pump, in the discharge main, with of course due consideration

of the conditions of entrance into the pump. For pumps of size over 8 inches, plainly made, and with heads ranging from 50 to 150 feet, efficiencies beyond 80 per cent are not uncommon. Fourteen-inch pumps, under such head conditions, are on record with 87½ per cent. It is therefore doubtful if any special features added to such pump casings could have increased the efficiencies.

The diffuser passages and particularly the diffuser vanes are deemed essential to secure the proper transformation of the kinetic energy of the water into potential. The kinetic head, as figured in preceding Chapters, amounts to a little over one-half of the total head; with an efficiency of 80 per cent it is easy to establish the fact that the loss is caused by leakage, friction of the impeller, a little friction of the delivered water, and the bearing friction. The question of guiding diffuser vanes may be easily disposed of with the statement, admitted by all, that they might be effective through a small portion of the delivery range of one given pump; therefore they would be detrimental to the rest of that range.

In a plain pump, whether of the volute type or not, the discharge end of the impeller appears, in a cross section of the apparatus, as a nozzle plunged into a mass of fluid, Figure 22; then, just as with steam or air, the flow is regular when the ratio $\frac{H_e}{H}$ is the proper one. H being the total head in the casing, and H_e the centrifugal head at the impeller outlet.

Judging by the two examples calculated before this ratio is less than 50 per cent. It would not help matters to increase it to the required 53 or 58 per cent, as for elastic fluids, because that would satisfy only the part of the range in the immediate vicinity of the maximum point. The issuing jet quickly spreads out and the transformation of the kinetic energy is even more rapid than the radial passage from R_1 to R_e. The straight part of the jet, close to the periphery is under the internal static pressure H_e, but it is pressed laterally by the surrounding fluid

under the full head H. It follows that a contraction should take place at the periphery, thus affecting the volume discharged and the pressure arrangement within the outer end of the impeller. This contraction actually occurs with "shallow" impellers, as described in the example of the preceding Chapter, and particularly so when the inlet edge of vane is wrongly designed.

The question is asked at once, if such is the case, why is there not ample evidence of trouble with many pumping units steadily in service? The answer is a comparatively easy one: the water in the discharge casing, being in contact with a relatively large rotating surface on each side of the impeller, acquires a certain degree of rotation and the mass thus forms a sort of protecting wall on each side of the jet. This is particularly true with large $\frac{R_o}{R_1}$ ratios.

It is interesting to state that in a goodly number of instances a certain size impeller has been found to fit in with an otherwise indifferent pump case, in such a manner as to produce the maximum obtainable results, both in efficiency and sustained performance. Such lucky combination was, of course, seized upon by observant designers, and studied in all its angles with a view to complete generalization applied to the various sizes manufactured by them. Unfortunately it is not of record that the lucky occurrence spread to adjoining fields.

From the foregoing the logical deduction is that a protection should be afforded to the peripheral jet; furthermore it should, for a given pump, be extended to all the possible sizes of impellers intended for that pump. A standard casing is usually designed to accommodate a certain range of impeller diameters; thus it might be said that a casing may receive impellers from 14 inches to 20 inches outside diameter. Now, if a working ratio $\frac{c_1}{c_o}$ is adopted as a practical standard, it is easy to outline a sufficient number of peripheral widths, and to draw curves tangent to the outer edges, Figure 24; then the

enclosing casing walls may be made practically parallel to these curves, and at such a distance from them as to allow ample freedom of rotation of the wheel, even when designed with a somewhat different $\frac{c_1}{c_0}$ ratio. The volute may then be located as shown. This insures a smaller bulk of the apparatus, hence a reduced weight and consequently cost. Besides, such a construction is inherently strong and does not require ribbing.

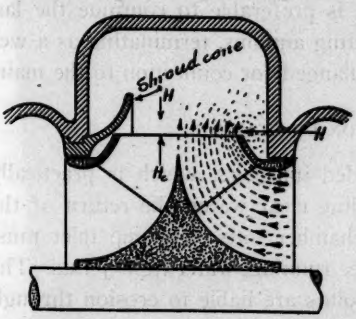


Fig. 23

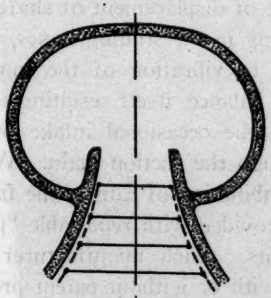


Fig. 24

Fig. 25

The same type of design is advocated for multistage pumps, with the difference, however, that the outer passage from one stage to the next has no volute, consisting merely of an easy but narrow bend. Assuming the velocity of entrance into the pump proper as 10 feet per second, the passage may be designed for a velocity of about 7 feet, depending of course upon the size of the unit, particularly as regards the core work in the foundry. The last stage may terminate in a volute, but that is not at all necessary, and it is preferable to continue the last bend into a concentric collecting annulus, terminating as a well proportioned delivery cone, flanged for connection to the main.

PREVENTION OF LEAKAGE.

In all pumps with shrouded impellers, which is practically the universal type for handling clear water, the return of the water from the discharge chamber into the pump inlet must be prevented by means of free running, water tight joints. The working surfaces of these joints are liable to erosion through the mere passage of water; to abrasion through the entrained particles of grit carried by the water; to actual wear through contact with one another because of displacement of shaft axis due to deformation of the packing in the stuffing boxes, or to wear in the main bearings, or to vibration of the impeller caused by slight loss of running balance, itself resulting either from actual unbalance or from the occasional intake of air from the stuffing boxes or through the suction main. Whatever may be the cause or the combination of causes, the fact is that all high grade pumps are provided with renewable "packing rings" at the running joints. Each manufacturer has adopted his own type of rings, with or without patent protection. The state of the art is so old in this respect that it is most improbable that any patent could hold in litigation, except as regards some special detail of construction.

The joints may be classified either as straight or as labyrinthine; and each of these types either as axial or as radial.

The four types are sketched in Figure 25, the pump axis being in all cases horizontal; the remarks in the preceding paragraph apply particularly to A and C. A may be considered as a very free running joint, since axial displacement is not limited by abutting surfaces. Whereas for B, C and D a thrust bearing must be used in order to maintain the impeller within fairly narrow limits of axial displacement. It will be noted that with A, B and C the water finally escaping from the joint impinges normally against the incoming suction flow, thereby causing some disturbance. With D the leakage flow mingles with the other with the least appreciable disturbance. Joints B and D are free from effects of unbalancing; and in D, while the radial clearances may be accordingly very large, the axial clearance can be maintained very close.

The question of the best type of running joint has caused much controversy and is likely to continue so doing, depending on standards of competition. The writer has had experience mainly with the three forms A, C and D. While using A for a long time as a standard many complaints arose from plants situated in various steel mills districts, because it was found necessary, on account of the particularly gritty and slightly acid conditions of the river water handled, to renew the packing rings about every month in order to keep the delivery of the pumps fairly within the requirements. At first the owners bought the repair parts from the manufacturers; they soon found this onerous beyond reason and then made the parts themselves. But they did not relish the frequent interruption of operations for the single purpose of renewing the rings; they came to the conclusion that the straight faced joint was deficient, and they decided to try another style, with a break in it, E, Figure 25. The results were gratifying because the life of the joint was extended to practically three months. The writer tried such a form for a while and then introduced the type C which, used under the same hard conditions mentioned above, proved to have a much better lasting power, practically

twice that of E. The writer finally adopted the type D, because in his estimation it had many points of superiority over the others.

AMOUNT OF LEAKAGE.

The flow through a restricted passage is usually calculated in the same manner as if it took place through a thin edge orifice, by means of the formula: $Q = v \times a \times K$; in which

Q = quantity in unit time,

v = velocity in unit time,

a = area of orifice,

K = coefficient of contraction = about 0.63.

Judging from very reliable tests it seems that the coefficient K has a higher value than 0.63; in fact, because the entrance edges are always somewhat rounded, the value is close to unity.

Let us assume that we deal with a straight joint of the form A in an ordinary pump casing of the multistage or single suction type, with a stage pressure of 125 feet. For comparison let us consider two sizes, viz: 3 inches, with a capacity range from 130 to 240 gallons per minute, and 8 inches, with a range from 900 to 1900 gallons per minute. The diameter of the running joint being 5¼ inches and 10¼ inches, respectively; the clearance space = 1/64-inch in each case.

$$v = \sqrt{2\,g\,H} = \sqrt{64.36 \times 125} \text{ feet per second.}$$

$$a = \pi D \times \frac{1}{64}$$

In gallons per minute we have, with two joints for each impeller:

$$2\left[60 \times \pi \times 5\tfrac{1}{4} \times \frac{1}{64} \times \frac{1}{231} \times 12 \times \sqrt{64.36 \times 125}\right]$$

$$= Q_3 = 45.6 \text{ gallons per minute}$$

$$2\left[60 \times \pi \times 10\tfrac{1}{4} \times \frac{1}{64} \times \frac{1}{231} \times 12 \times \sqrt{64.36 \times 125}\right]$$

$$= Q_8 = 88 \text{ gallons per minute}$$

At the extreme ranges:

$\frac{45.6}{240}$ = 19 per cent, with K = 1 or 12 per cent with K = 0.63.

$\frac{88}{1900}$ = 4.635 per cent with K = 1 or 2.92 per cent with K = 0.63.

To these losses must be added, of course, the leakage around the shaft, that is, from one stage to the preceding one. This may be calculated in the same manner. But leaving this point aside, we may easily draw important conclusions: (*a*) leakage losses are greater for small pumps than for large ones; (*b*) consequently the efficiency of small pumps cannot be otherwise than below that of the larger ones: (*c*) a double suction, single impeller pump is more efficient than one impeller of a multistage type, for pumps of the same size and under precisely the same operating conditions. These points will undoubtedly be found interesting.

Unfortunately, after a certain amount of service, wear is bound to take place, and the clearance space is increased with consequent increase of leakage losses and decreased efficiency. An ordinary wear of $\frac{1}{64}$-inch will double the above figures.

With a well designed pump casing, along the lines of Figure 24, the head on the joint is bound to be smaller than that obtaining in the outer discharge chamber, hence Q will be reduced appreciably, and will be the more so if the pump has efficient labyrinthine joints.

PART VI.—PERFORMANCE CURVE—DIAMETER OF IMPELLER.

A quantity of water Q, in gallons per minute, is required to be delivered at a head or pressure H, expressed in feet. A centrifugal type of pump being agreed upon, the quantity Q determines its size, understood usually as the diameter of the inlet opening. Each pump size is rated for a predetermined useful range of delivery, any point of which may be selected

to establish the point of maximum efficiency, or in shop parlance: the *normal point,* or the *point.* A short table is given here which agrees with the best practice.

Size	3"	4"	5"	6"	8"	10"	12"	14"	16"	18"	20"
Min. g.p.m.	130	240	370	600	900	1500	2500	3500	5000	7000	9000
Max. g.p.m.	240	440	700	1000	1900	3000	4500	6200	8100	10500	13000

In the absence of such table the following formula may be used, based on a velocity $C_p = 10$ feet per second.

$$D_p = 0.64\sqrt{\frac{Q}{C_p}} = 0.202\sqrt{Q}.$$

Thus, assume $Q = 3000$ gallons per minute, then $D_p = 11.06$ inches. But 11-inch pumps are not in use, therefore we have a choice between a 10-inch or a 12-inch size, with $C_p = 12.25$ feet and $C_p = 8.5$ feet, respectively, the final decision depending upon other operating conditions, or upon cost, the smaller unit being of course less expensive.

With the two factors Q and H we have the water HP, by means of the formula: $\text{WHP} = \frac{Q \times H}{3960}$, which is commonly used and is based upon 8.34 pounds as the weight of the U. S. gallon of clear water.

Depending upon the size of the pump we may assume a convenient efficiency. This is simply to obtain an idea of the B.H.P. required at the point. The following table is given for the purpose of furnishing a rough estimate of the efficiency when the impeller has a $\frac{R_o}{R_1}$ ratio of *about 2;* as such it gives a fair indication, useful for the purpose, but it should never be used for definite work, that is, for guaranteed performance.

Gallons per minute	120	200	350	550	950	1600	2600	3500
Efficiency, per cent	48	57	62	66	69	72	74	76

Let the conditions warrant an efficiency E = 72 per cent, then B.H.P. = $\frac{\text{W.H.P.}}{0.72}$. Past the *point* the B.H.P. may be expected to increase somewhat, depending upon the impeller design. We can thus establish a rough estimate of the power required for the prime mover, which leads directly to the determination of the speed of rotation N = R.P.M.

The next step is to ascertain what other conditions are to be met, such for instance as a lesser Q at a higher head. Several such conditions may be desired. When all this information is at hand a diagram may be made on cross section paper; the Q are plotted against the heads, and the points thus determined are connected by a roughly drawn curve. If the latter is fairly regular it may be extended to the origin, where Q = zero, the ordinate of which becomes the *shut off pressure.* If the curve is not fair, one may be drawn which will insure the additional conditions with some minor changes, and will meet exactly the original, or *main* QH condition specified. This presumes, of course, that the designer has experience in this line of work; if he has not he may determine a few points by very simple computation, as will presently be shown, and these, when connected, will give a curve of sufficient accuracy, particularly so on each side of the point, thus covering the most useful part of the range; that will be the performance curve [estimated]. The true performance curve is obtained from the shop tests, where the pump is run at a *constant rate of speed,* and the delivery is decreased by steps from Q to zero, the corresponding heads being recorded carefully, after which the Q are plotted against the H on the diagram. Throughout this discussion the speed is assumed constant.

The point most important at present is the *Shut off Pressure* H_s, at which the delivery valve is entirely closed, and no water leaves the pump. Unfortunately it is absolutely impossible to either measure or calculate such a pressure exactly, because several factors seriously interfere with its steadiness. In the first

place we do not know the size of the disk of water rotated by the impeller, hence the diameters of the various concentric layers, and consequently the total centrifugal pressure generated. We are positive about the annulus R_1 to R_o within the impeller, but there is a mass of water entrained by the outer periphery and another one by the inner periphery. Then, as the H_s is usually much greater than the head at the point, the leakage loss is a more important item, and there is a constant flow from the discharge chamber into the impeller intake. The presence of small quantities of air or aqueous vapor constantly dislodged from one part to another is a source of unsteadiness. As a result of such conditions the pressure gauge is rather jumpy, vibrating through a range of 10 to 20 feet, and sometimes more in extreme cases, so that the exact head cannot be read off. However, it is practically at the midpoint of that range, and for all ordinary purposes its value is given by the equation.

$H_s = \frac{V_o^2}{2g} \times K$, V_o being the peripheral impeller velocity, in feet per second. Some designers use a value of K varying from 0.75, for $\frac{R_o}{R_1} = 1.5$, to unity for $\frac{R_o}{R_1} = 2.25$ and over. This may hold well enough for a certain type of pump, but the writer observed that K may be dispensed with in general practice, reducing the formula to: $H_s = \frac{V_o^2}{2g}$.

$$V_o = \frac{\pi D_o}{12} \times \frac{N}{60} = \text{feet per second.}$$

$$\therefore D_o = \sqrt{2gH_s} \times \frac{720}{\pi N} = 1838.61 \frac{\sqrt{H_s}}{N}.$$

The numerical coefficient is rounded up to 1840, so that finally:

$$D_o = 1840 \frac{\sqrt{H_s}}{N}$$

With the outer diameter thus obtained all the factors necessary for the design of the impeller are at hand.

However, perhaps in the majority of cases no other conditions but Q and H are specified and the form of the performance curve is left entirely to the judgment of the designer. It is usual to state that three forms are available, one most commonly employed, like A, Figure 26, with a gentle slope from H_s to H; a second, B, rising from H_s to H, and a third one, C, running practically horizontally from H_s to H. Of course, steeper curves like D and E may be required at times, for instance, when a high speed must be used and the impeller would otherwise be too shallow. With A, D, E, the shape of the vane is always convex in the direction of rotation; with C the vane has a double curvature, being convex in the first part, from R_1 to a certain point R, and then concave to R_o, with the last element practically radial; with B the vane is first convex, then concave, terminating at an acute angle with the circumference of the wheel. C is used for a uniform head with variable delivery, it requires some sort of governor to regulate the speed; B is not common in this country; it is used to some extent in Europe, the purpose being to obtain a high efficiency, through reduction of the outer diameter and consequently, as we will see further, of frictional work; it covers a relatively small useful range and is adaptable only to a limited set of operating conditions; besides, it has some serious disadvantages in the way of priming and unpriming. It is needless to say that any of these forms of impellers may be computed by means of the formulas developed in this work, the only difficulty—a small one—being in the determination of the radii of curvature of the vane, particularly in the neighborhood of the change of curvature.

For all ordinary purposes, then, a performance curve with a slope like A is strongly recommended. It is not so sensitive to increased leakage due to wear of the rings, nor to slight variations of speed likely to affect any motor.

It may be remarked that the performance curve is also called the PV, or pressure-volume curve.

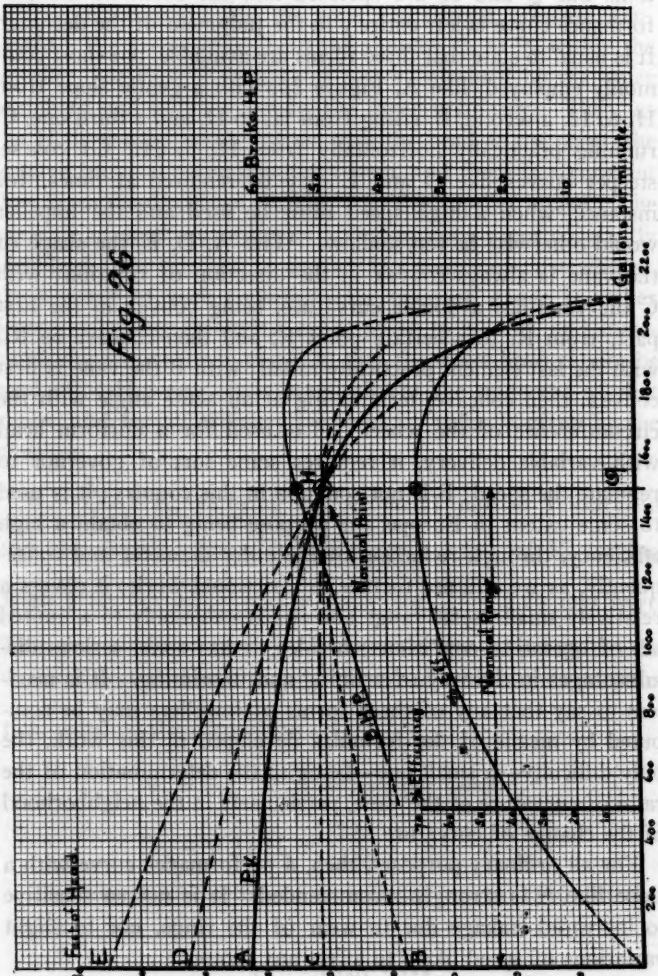


Fig. 26
H.P. Brake
Gallons per minute
Natural Point
Normal Point
Eff.
Feet of Head
E
D
A
C
B

In the Figure the efficiency and B.H.P. curves have been drawn to suit curve A.

DELINEATION OF THE PV CURVE.

The full range of the pump extends from zero to the maximum delivery at zero head; it is divided into the *normal range,* extending from zero to Q, and the *post normal range,* from Q to the maximum. The best way to proceed is perhaps to divide the normal range into several equal parts, say three, for instance. In Figure 26, we would have then: $Q_1 = 500$ gallons per minute, $Q_2 = 1000$ gallons per minute; with these the corresponding radial velocities are obtained. The impeller has been fully determined and the factors needed are at hand, they are the a and the R, and now the ω may be obtained graphically for each set of R. After which the ω^2 may be plotted on a diagram and by graphical integration—as explained previously in the text—the value of the total centrifugal pressure $\int \frac{\omega^2}{g} \frac{dR}{R}$ is very readily obtained. This is done for Q_1 and for Q_2. In each case, of course, the kinetic head is obtained by the formula $H_k = \frac{\Omega^2}{2g}$, so that finally we have $H_1 = H_{c1} + H_{k1}$, and $H_2 = H_{c2} + H_{k2}$. The plottings Q_1H_1 and Q_2H_2 give two points of the PV curve, making with H_s and H four points in all, which permits a satisfactory delineation of the PV curve for the normal range.

It may be objected that no mention has been made of shock at the entrance of water into the vane space. This is true, and there may be some effect on the final pressure as a result from this shock, but there is no known way of satisfactorily computing this shock and its effects, and after all it seems probable that in this part of the range the water is entrained to some extent in the vicinity of the inner periphery and that it does not enter the vane space in a true radial direction as is the case

at the normal point. This view of the matter is somewhat mitigated by the fact that the computed plottings agree very fairly with those obtained from actual test.

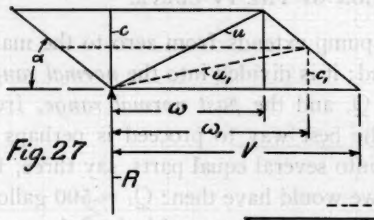


Fig. 27

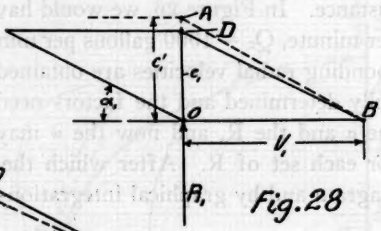


Fig. 28

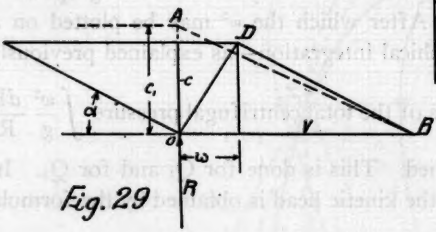


Fig. 29

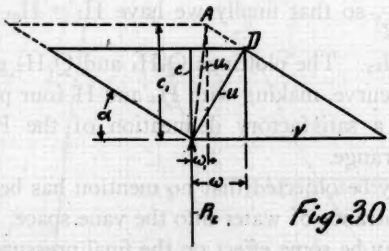


Fig. 30

The ω mentioned above are obtained from a velocity diagram drawn for each R of a given set. In Figure 27, the diagram in full lines represents the various velocities and the vane angle α for one of the R, corresponding to operation at the normal point with the radial velocity = c. Now, for a reduced

volume Q_1, for instance, c is reduced to c_1, whence u becomes u_1 and ω becomes ω_1. The process is very simple. It is this ω_1 which is used for the graphical integration. It is obvious that the resulting head H_1 is bound to be greater than H since ω_1 is greater than ω.

POST NORMAL PV.

Here we meet with difficulty at the start, for, in actual test it is not possible to measure the ultimate point of the range on account of the disturbances in both the flow and the pressure gauges. Therefore it is necessary to proceed step by step towards that ultimate point. In the case illustrated in Figure 26, $Q_3 = 1700$ gallons per minute, and $Q_4 = 1900$ gallons per minute may be selected. The velocity diagrams must be considered first because, here, the conditions are quite different from those of the normal range. In Figure 28, the diagram for the first element of the vane, for radius R_o is in full lines. Draw AO = c_1 = radial entrance velocity for Q_3; now if we were to join A and B by a straight line, this latter would not coincide, in position, with DB = w = relative velocity. AB is then said to be outside of DB. *This shows that the first element of the vane is without action upon the entering stream.* It is then necessary to try one R after another until a suitable diagram is found. Thus, in Figure 29, AB is inside of DB *and does not coincide with it.* However, the closeness of the lines indicates that the final R tried is beyond the first point *of effective contact of the vane with the entering stream.* A diagram may now be established, intermediate between this R and the preceding one, and the exact point of entrance contact fairly well determined; call this R_e. It should be noted that the water entered the impeller at R_o with the radial velocity c_1, has traversed a vane canal of varying sectional area, and has reached a point R_e where it is being acted upon, its actual velocity then being in accordance with the scale of the $\frac{c_1}{c_o}$ ratio selected for the impeller.

This point once established ω is obtained for each of the remaining R, from the corresponding velocity diagram such as shown in Figure 30, all the values of ω are then plotted as before, and the graphical integration performed, which gives H_c. This added to $H_k = \frac{\Omega^2}{2g}$, gives H_3, or total head for $Q_3 = 1700$ gallons per minute.

The same procedure is followed for H_4, H_5, etc., so that finally as many points of the post normal PV are secured as desired.

PV CONTROL.

In Figure 31, the PV curve extends to a maximum delivery of 2900 gallons per minute; it is assumed to represent fairly well the performance of an impeller in which the radial velocity is uniform, that is, with $\frac{c_1}{c_o} = 1$. Sometimes such a performance is required. It will be noted that the B.H.P. rises steadily from 61.5 at the normal point to a maximum of 77, or over 26 per cent above that at the point; this means that a large motor must be provided, or that if a smaller motor is used the extra power needed will have to be furnished by overloading the motor. The present case is relatively mild; in some installations the overload may be such that, when sustained, it may cause the motor to burn out. Many such instances are on record. To obviate this trouble the ratio $\frac{c_1}{c_o}$ is selected different from unity; as advocated before $\frac{c_1}{c_o} = \frac{10}{7}$ is perhaps the most serviceable. In the figure curve A is drawn for such a ratio, and curve B for $\frac{c_1}{c_o} = \frac{10}{5}$. The post normal range is thus much shortened in either case, and likewise the

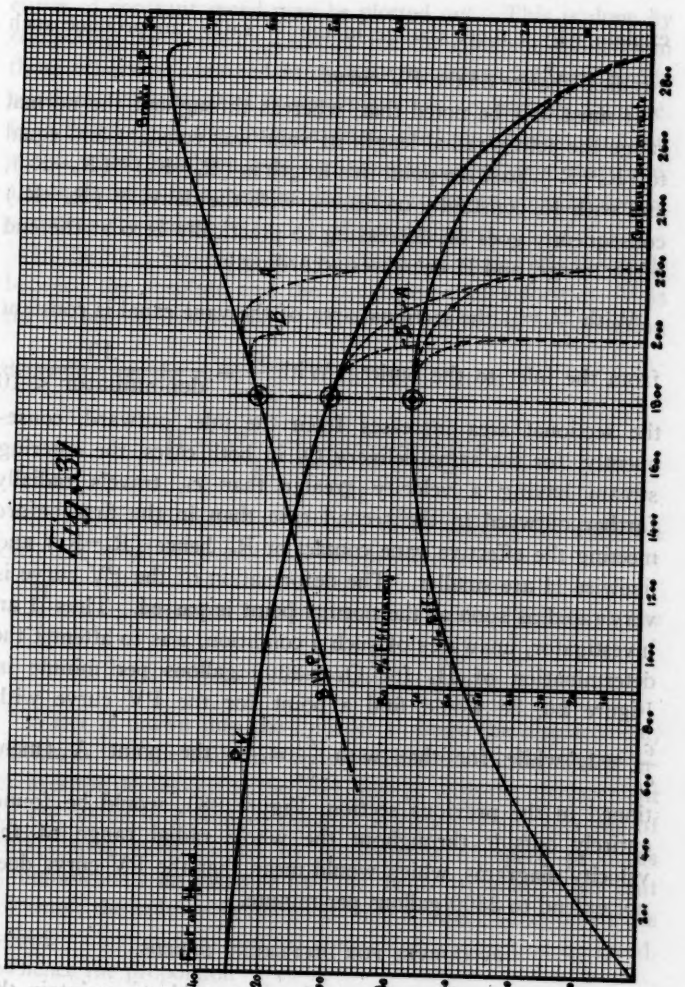


Fig. 31.
Feet of Water.
H.P.
Efficiency.
Revolutions per minute.

maximum B.H.P. very much reduced. It remains now to explain why a mere change in the $\frac{c_1}{c_o}$ ratio can produce such a radical change of the PV curve.

It must first be noted that, whereas throughout the normal range the acting radial length of vane was constant and equal to $(R_o - R_1)$, past the point this length is $(R_o - R_e) < (R_o - R_1)$, and with R_e constantly increasing with the delivery $(R_o - R_e)$ consequently goes on decreasing to practically zero at the end of the range, and H decreases accordingly.

With $\frac{c_1}{c_o} = 1$ the sectional area of the vane canal is constant from the inlet to the outlet. With $\frac{c_1}{c_o} = \frac{10}{\text{denominator} < 10}$ the sectional area increases from the inlet outward, consequently, for the same delivery Q_x in both cases, the entering stream having a velocity greater than c_1, travels radially farther outward in the second case than in the first before meeting the effective vane contact at R_e, hence $(R_o - R_e)$ and likewise H are smaller. The deformation of the PV curve is very rapid as soon as the normal point is passed. Thus if an investigator, unawares of such a condition, was to attempt the determination of H_x for $Q_x = 2300$ gallons per minute in Figure 26, he would find a point for the PV curve with $\frac{c_1}{c_o} = 1$, while for that with $\frac{c_1}{c_o} = \frac{10}{7}$ the point A (mentioned in the previous section, Figure 28), would be found entirely outside throughout the whole radial range of the velocity diagrams, which would mean that the PV curve does not extend to 2300 gallons per minute.

PV CURVES FOR DIFFERENT SPEEDS.

In testing a pump it is not always possible to maintain the speed constant, hence in taking readings for any set of Q_xH_x that of the corresponding speed is also accurately taken. Upon

completion of the test the readings are arranged so that a PV curve of constant speed may be plotted out. This is done by computing the needed Q and H for the constant speed from those otherwise obtained in the test, by means of very simple relations. The method is convenient but not strictly accurate. It is based upon the formula established by Combes, in 1840 about, which has been used as a standard ever since that time, although made to appear in different forms to suit the special purposes (or claims to fame) of the designers.

In the form presented here α is the vane angle; β is the angle between the absolute and the circumferential velocities; V is the circumferential velocity; and K is a coefficient of correction depending on the ratio $\frac{R_0}{R_1}$. With these factors:

$$H = \frac{V^2\left[1 + \frac{\sin(\alpha - \beta)}{\sin(\alpha + \beta)}\right]}{2g} . K.$$

It is quite obvious that for any other velocity V_x due to a different speed we would have the proportion:

$\frac{H}{H_x} = \frac{V^2}{V_x^2}$, in other words, the total pressure varies as the square of the speed of rotation, or $\frac{H}{H_x} = \frac{N^2}{N_x^2}$.

Water is drawn into the impeller through the displacement, outwards, of the mass of water within the latter, which increases the depression at the inlet, just as with the displacement of the piston in a plunger pump; consequently any increase of this displacement caused by an increase of speed produces simultaneously an increase of flow, hence of volume per unit of time; the volume delivered is then a direct function of the speed. This holds as well of course for a reduction of speed. Whence the proportion

$$\frac{Q}{Q_x} = \frac{V}{V_x} = \frac{N}{N_x}.$$

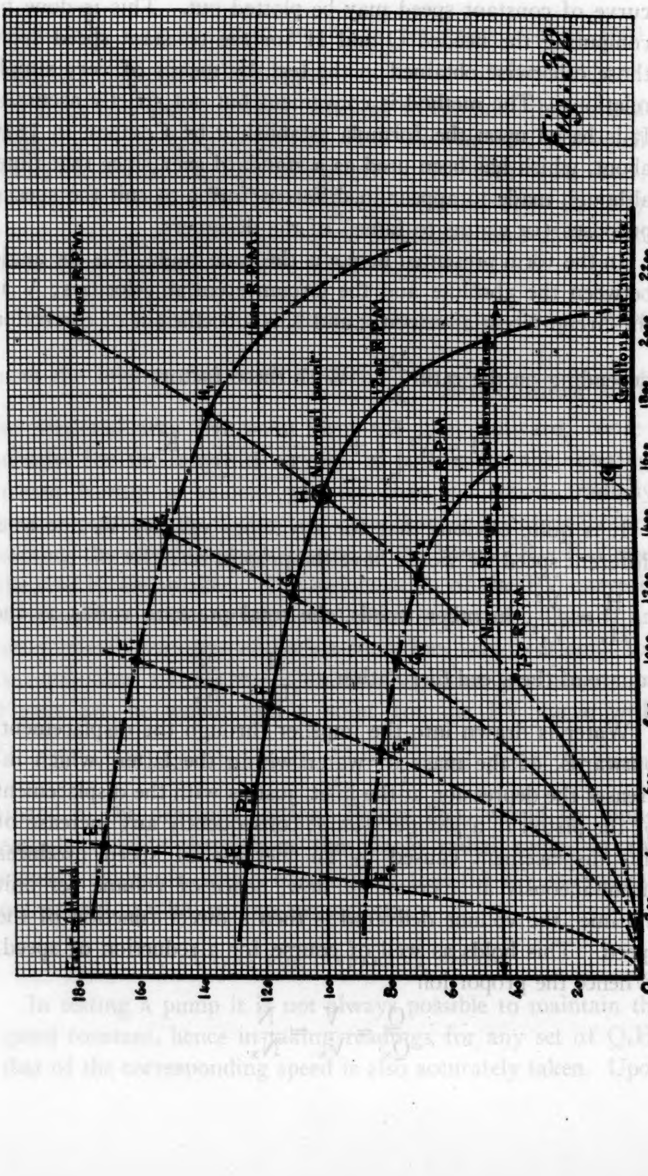


Fig. 32
Gallons per minute
Normal Point
Normal Range
1800 R.P.M.
1600 R.P.M.
1200 R.P.M.
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
20
40
60
80
100
120
140
160
180

On the diagram in Figure 32, the normal PV curve is shown in full line for a speed of 1200 R.P.M.; two other curves are in dotted lines, one for 1000, and the other for 1400 R.P.M. It will be noted that for one given point H the effects of a speed variation from zero to 1600 R.P.M., for instance, are plotted on a curve $O\ H_2\ H\ H_1$ which is a parabola. The process applies to any point of the normal curve, so that by drawing a series of parabolas the effects of any speed variation may be quickly plotted out. The computation of H_x and Q_x is most easily done with the slide rule.

REMARK.

It should be well kept in mind that the method is not accurate to the extent that a designer could guarantee with assurance that a new PV curve computed as above would be proof against test, except in case of small speed variation like 5 to 6 per cent on either side of the normal PV. However it proves useful in drawing up estimates, particularly for a pump intended primarily for ordinary service and secondarily for emergency, such as for fire or for general high pressure service.

A most important question with speed variation work is that of the B.H.P. involved. By combining the two formulas

$$\left(\frac{H}{H_x} = \frac{N^2}{N^2_x}\right) \text{and} \left(\frac{Q}{Q_x} = \frac{N}{N_x}\right)$$

we obtain:

$$\frac{Q \times H}{Q_x \times H_x} = \frac{V^2 \times V}{V^2_x \times V_x}, \text{ or } \frac{Q\,H}{Q_x\,H_x} = \frac{N^3}{N^3_x}.$$

This signifies that the water H.P. varies as the cube of the speed.

It has been observed in numerous tests that for any given point the efficiency remains practically constant throughout a goodly portion of the range of speed variation. Keeping in mind that here also strict accuracy is not guaranteed, we have

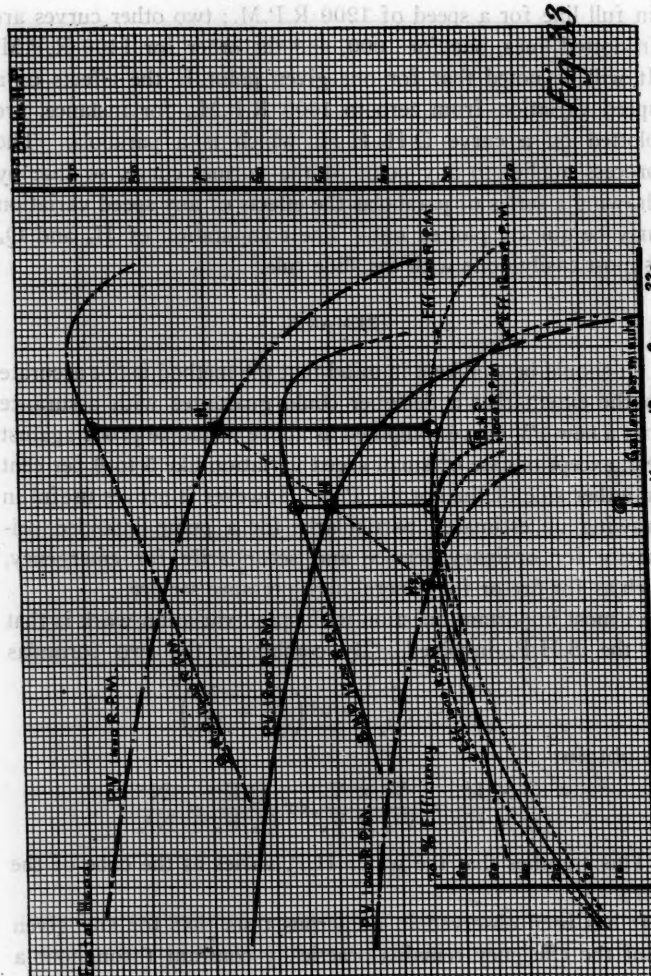


Fig. 33

nevertheless at hand a very useful means of computing the required B.H.P., because according to the above we have also:

$$\frac{\text{B.H.P.}}{\text{B.H.P.}_x} = \frac{N^3}{N^3_x}$$

In Figure 33, the efficiency and B.H.P. curves are shown for the conditions considered in the preceding case.

PUMP EFFICIENCY.

It does not seem possible to differentiate between the efficiency of the impeller proper and that of the pump as a whole, although this subject has tempted many writers. This should be obvious from the many points discussed in this work; from a correctly made impeller fitted into a well designed pump case an overall efficiency will be obtained greater than could be expected from the same impeller fitted into an improperly made pump case.

The general expression for the overall efficiency is:

$$e = \frac{\text{W.H.P.}}{\text{W.H.P.} + \text{losses}}.$$

In view of the importance of the efficiency factor in commercial transactions where the purchaser insists upon a rigid guaranty of performance, and also where keen competition requires guaranties of high grade service, it is well to modify this expression so as to derive from it formulas in accordance with the best practice.

It is not possible to use a "global" coefficient to replace the "losses" factor. The losses are of two kinds: frictional work and leakage. Fortunately part of the frictional H.P. may be expressed in concise form. It depends on the outer Diameter D_o and the revolutions per minute or N. So that we may write:

$$\text{Frictional H.P.} = \text{F.H.P.} = 0.038 \left(\frac{D_o}{10}\right)^4 \left(\frac{N}{1000}\right)^3.$$

If D_o is replaced by its value: $\frac{1840\sqrt{H_s}}{N}$, we have also:

F.H.P. $= 0.435 \frac{H_s^2}{N}$, which is a more convenient form.

By means of these formulas straight line, logarithmic nomograms may be established which will be found most invaluable time savers in regular pump work. Figure 34 is such a nomogram; its main defect is that it is drawn too small; besides, it should cover a wider field. However, it is submitted here merely as an example. Most everyone is familiar with such charts, so that a few words of demonstration will suffice. As an instance, take the case illustrated in Figure 33, where Q = 1500, H = 100 feet, H_s = 130 feet, and N = 1200. By placing a straight line from 130 = H_s on the first line, to 1200 = N on the fourth line, the intersection on the second line will give F.H.P. = 6.15, and on the third line D_o = 17.48 inches.

Right here we should remark that a more rounded figure for D_o may be desirable for impeller calculations proper; this can be obtained by slightly shifting up or down the H_s value.

Since leakage cannot be prevented it is imperative from the start to make allowance for it. Thus if a delivery of 1000 gallons per minute is specified, the impeller should be designed to suit a larger quantity; the margin may range from 5 to 10 per cent, according to conditions and to the experience acquired. Likewise a slightly greater head should be used, as some of it is bound to be lost along with the leakage. It is difficult to advise properly a newcomer in the field about the value of such margins of safety. The best way is to so select a normal point that the test curve will pass just outside of that specified. For an ordinary case an increase of 3 to 5 per cent in the H and 5 to 10 per cent in the quantity are good figures for a start, and they may be reduced advantageously as experience is gained.

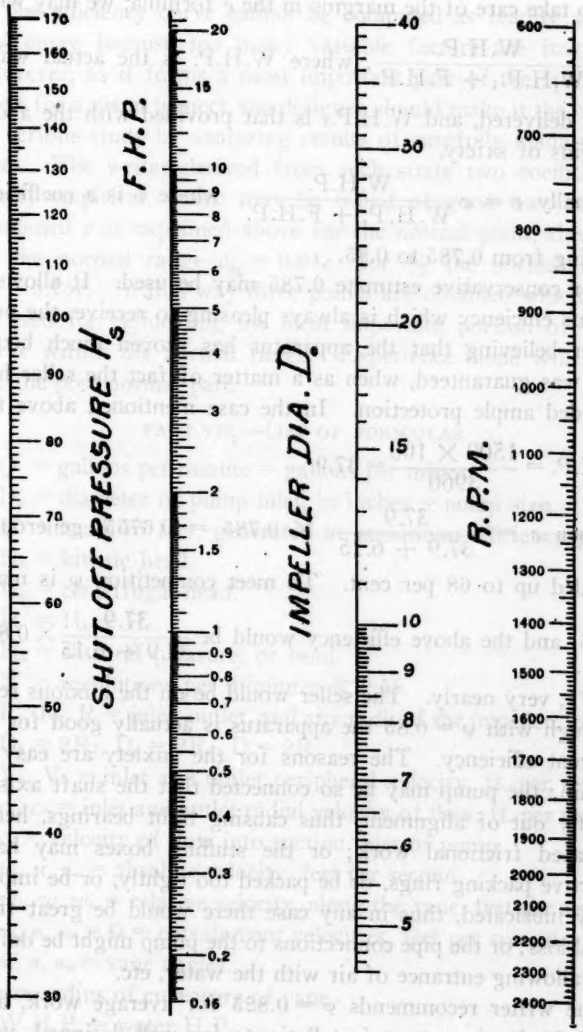

SHUT OFF PRESSURE H_s
F.H.P.
IMPELLER DIA. D_e
R.P.M.
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
15
10
9
8
7
6
5
4
3
2
1.5
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.16
40
30
20
15
10
9
8
7
6
5
600
700
800
900
1000
1100
1200
1300
1400
1500
1600
1700
1800
1900
2000
2100
2200
2300
2400

FIGURE 34.

To take care of the margins in the e formula, we may write $e = \frac{\text{W.H.P.}}{\text{W.H.P.}_1 + \text{F.H.P.}'}$, where W.H.P. is the actual water H.P. delivered, and W.H.P.$_1$ is that provided with the above margins of safety.

Finally $e = \varphi \frac{\text{W.H.P.}}{\text{W.H.P.} + \text{F.H.P.}'}$, where φ is a coefficient ranging from 0.785 to 0.85.

For conservative estimate 0.785 may be used. It allows a surplus efficiency which is always pleasing to receive, the purchaser believing that the apparatus has proven much better than was guaranteed, when as a matter of fact the seller had provided ample protection. In the case mentioned above the

$$\text{W.H.P.} = \frac{1500 \times 100}{3960} = 37.9.$$

Then $e = \frac{37.9}{37.9 + 6.15} \times 0.785 = 0.6755$, generously rounded up to 68 per cent. To meet competition φ is made 0.825, and the above efficiency would be $\frac{37.9}{37.9 + 6.15} \times 0.825 = 0.71$, very nearly. The seller would be on the anxious seat, although with $\varphi = 0.85$ the apparatus is actually good for 73 per cent efficiency. The reasons for the anxiety are easy to explain: the pump may be so connected that the shaft axis is slightly out of alignment, thus causing tight bearings, hence increased frictional work; or the stuffing boxes may have defective packing rings, or be packed too tightly, or be imperfectly lubricated, thus in any case there would be great frictional loss; or the pipe connections to the pump might be defective, allowing entrance of air with the water, etc.

The writer recommends $\varphi = 0.825$ for average work, and $\varphi = 0.85$ for important installations; for well designed, very large units, $\varphi = 0.90$ may be confidently employed.

The efficiency curve cannot be computed as readily as the PV curve because too many variable factors are involved. However, as it forms a most important part of the specifications for a given project, the designer should make it the object of serious study by analyzing results of carefully made pump tests. The writer derived from such study two coefficients, purely empirical, which may be found of good use: having computed e as explained above for the normal point, then for ¾ the normal range $e_{¾} = 0.94e$; for ½ the normal range $e_{½} = 0.75e$. In this way three points are obtained which may be used for delineating the most important portion of the e curve within the normal range. Experience alone will serve for the post normal part.

PART VII.—LIST OF FORMULAS.

Q = gallons per minute = gallons per minute.
D_p = diameter of pump inlet, in inches = pump size.
H = head, in feet, generated at maximum efficiency point.
H_k = kinetic head.
H_c = centrifugal head.
$H = H_k + H_c$.
H_s = shut off pressure, or head.
N = revolutions per minute = R.P.M.
R_1, R_o, R = inlet, outlet, and any radii of the impeller, inches.
$D_1 = 2R_1$, $D_o = 2R_o$, $D = 2R$.
V_1, V_o = inlet and outlet peripheral velocity, ft. per second.
c_1, c_o = inlet and outlet radial velocity of flow, ft. per second.
c_p = velocity of flow into suction inlet of pump.
u_1, u, u_o = absolute velocity, feet per second.
w_1, w, w_o = relative velocity, along the vane, feet per second.
ω_1, ω, $\omega_o = \Omega$ = entrainment velocities, feet per second.
a_1, a, a_o = vane angles.
ρ = radius of curvature of vane.
W.H.P. = water H.P.
B.H.P. = brake or shaft H.P.
F.H.P. = frictional H.P.
e = efficiency.

$$D_p = 0.64\sqrt{\frac{Q}{c_p}}.$$

To be rounded up to suit standard pipe sizes.

For preliminary estimate assume $c_p = 10$ feet, then $D_p =$

$$0.202\sqrt{Q}. \quad c_p. = 0.408\frac{Q}{D^2_p}$$

It is advantageous to make c_1 greater than c_p, it insures a more compact flow into the impeller inlet. In good practice: $c_1 = 1.1\ c_p$ about, so that c_1 is conveniently rounded up.

$\frac{c_1}{c_o}$ = ratio of inlet to outlet radial velocities; $\frac{c_1}{c_o} = \frac{10}{7}$ recommended.

W.H.P. $= \frac{Q \times H}{3960}$. For estimating purposes the diagram on Figure 37 may be used.

Assume e for preliminary estimate.

$\therefore$ B.H.P. $= \frac{\text{W.H.P.}}{e}$. This gives an idea of the size of motor required and of the available speeds.

The speed should not be too great for the size of pump, otherwise the inlet vane angle a_1 would be undesirably small. Satisfactory limiting values are:

$N = 420\frac{Q}{D_p{}^2}$, for single suction, with $a_1 = 13° 46'$.

$N = 590\frac{Q}{D_p{}^2}$, for double suction, with $a_1 = 13° 59'$.

H_s = Shut off pressure, is selected to suit specifications, or according to the judgment of the designer.

D_o = Impeller diameter, in inches, $= \frac{1840\sqrt{H_s}}{N}$.

Size to be rounded up a little by slight change of H_s.

N must be such that a suitable radial vane length ($R_o - R_1$) is secured, hence if N is fixed, it may be necessary to increase H_s; this will decrease the efficiency.

So far H_s has been understood as the shut off pressure for one impeller; it will remain so. The total head H may be such that it cannot be generated by one single stage, therefore it must be divided by the number of stages selected according to specified conditions and to facilities at hand.

c = radial velocity of flow for radius R; $c = c_1 - (c_1 - c_o)\frac{h}{H}$.

Divide H into 10 equal parts, then $\frac{h}{H} = 0.1, 0.2, 0.3, \cdots$ etc.; correspondingly $c = \left[c_1 - (c_1 - c_o) \frac{h}{H} \right] = c_1$ $[1 - (1 - b)\ (0.1,\ 0.2,\ 0.3, \cdots \text{etc.})]$, with $b = 0.9, 0.8, 0.7, 0.6, 0.5$, corresponding to $\frac{c_1}{c_o} = \frac{10}{9}, \frac{10}{8}, \frac{10}{7}, \frac{10}{6}, \frac{10}{5}$.

Let K_c = coefficient for c, depending on $\frac{h}{H}$ and $\frac{c_1}{c_o}$, then $c = K_c\, c_1$.

K_c may be obtained from the following table.

R is obtained from $\frac{R - R_1}{c_1 + c} \cdot \frac{c_1 + c_o}{R_o - R_1} = \frac{h}{H}$.

Let K_R = a coefficient for R, depending on $\frac{h}{H}$ and $\frac{c_1}{c_o}$; it is given in the following table along with K_c.

$$R = K_R (R_o - R_1) + R_1$$

$$\left(\frac{R}{R_1} - 1\right) = (n - 1) = K_R \left(\frac{R_o}{R_1} - 1\right).$$

With this value of $(n - 1)$ determined for each consecutive h, the coefficient K_ω is obtained from one of the Charts, and the corresponding $\omega = K_\omega \sqrt{h}$. [See text for complete equation.]

	K_c					K_s				
$\frac{b}{H}$	$\frac{c_1}{c_e} = \frac{10}{9}$	$\frac{10}{8}$	$\frac{10}{7}$	$\frac{10}{6}$	$\frac{10}{5}$	$\frac{10}{9}$	$\frac{10}{8}$	$\frac{10}{7}$	$\frac{10}{6}$	$\frac{10}{5}$
0.1	0.99	0.98	0.97	0.96	0.95	.10473	.11000	.11588	.12250	.13000
0.2	0.98	0.96	0.94	0.92	0.90	.20842	.21777	.22824	.24000	.25333
0.3	0.97	0.94	0.91	0.88	0.85	.31105	.32333	.33706	.35250	.37000
0.4	0.96	0.92	0.88	0.84	0.80	.41474	.42666	.44235	.46000	.48000
0.5	0.95	0.90	0.85	0.80	0.75	.51815	.52777	.54412	.52250	.58333
0.6	0.94	0.88	0.82	0.76	0.70	.61263	.62666	.64204	.66000	.68000
0.7	0.93	0.86	0.79	0.72	0.65	.71106	.72333	.73706	.75250	.77000
0.8	0.92	0.84	0.76	0.68	0.60	.80842	.81777	.82824	.84000	.85333
0.9	0.91	0.82	0.73	0.64	0.55	.90472	.90999	.91588	.92250	.93000
1	0.90	0.80	0.70	0.60	0.50	1.	1.	1.	1.	1.

The circumferential velocity $V = \frac{D\,N}{229.18}$.

The vane angle α is obtained from trigonometric tables with

$$\text{tang } \alpha = \frac{c}{V - \omega}.$$

The radius of curvature for the arc of vane intersecting two consecutive circles R and R′ is

$$\rho = \frac{R^2 - R'^2}{2\,[R\cos\alpha - R'\cos\alpha']}$$

Axial width of vane:

$$b = \frac{0.32084\,Q}{c\,(2\,\pi\,R - n\,t_v)},$$

t_v is the "circular" thickness of vane, for radius R, n is the number of vanes, so that nt_v is the total "circular" thickness.

Efficiency at the normal point:

$$e = \varphi\,\frac{W.H.P.}{W.H.P. + F.H.P.}$$

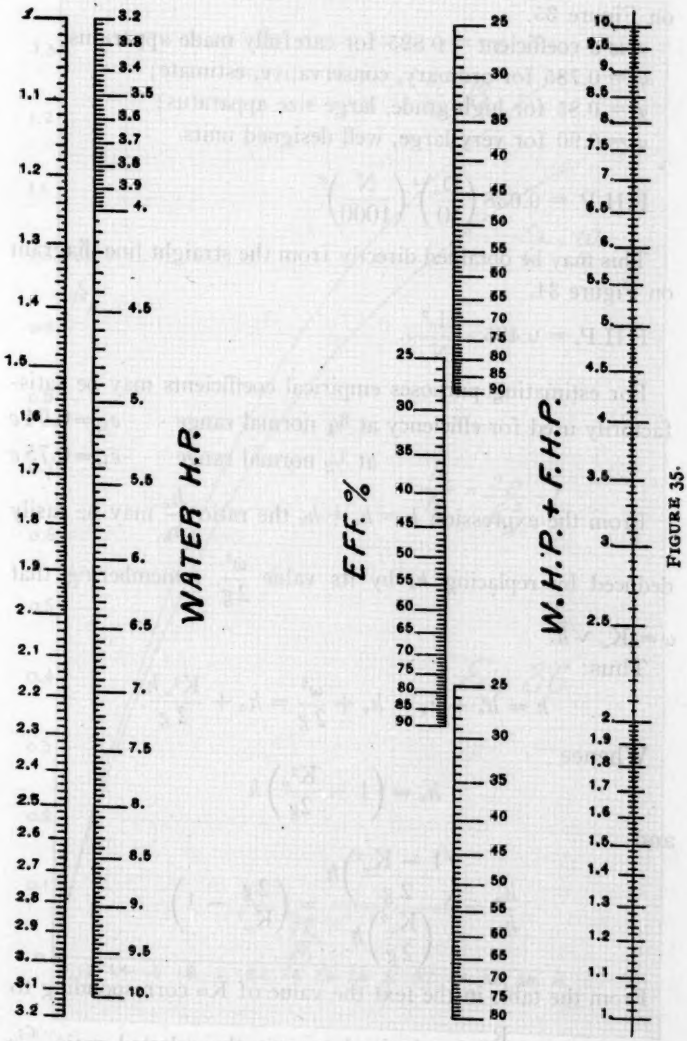


FIGURE 35.

This may be obtained directly from the straight line diagram on Figure 35.

φ is a coefficient = 0.825 for carefully made apparatus;
φ = 0.785 for ordinary, conservative, estimate;
φ = 0.85 for high grade, large size apparatus;
φ = 0.90 for very large, well designed units.

$$\text{F.H.P.} = 0.038 \left(\frac{D_o}{10}\right)^4 \left(\frac{N}{1000}\right)^3$$

This may be obtained directly from the straight line diagram on Figure 34.

$$\text{F.H.P.} = 0.435 \frac{H_s^2}{N}.$$

For estimating purposes empirical coefficients may be satisfactorily used for efficiency at ¾ normal range $\cdots e_{¾} = 0.94\, e$
at ½ normal range $\cdots e_{½} = 0.75\, e$

From the expression $h = h_c + h_k$ the ratio $\frac{h_c}{h_k}$ may be easily deduced by replacing h_k by its value $\frac{\omega^2}{2g}$, remembering that $\omega = K_\omega \sqrt{h}$.

Thus:

$$h = h_c + h_k = h_c + \frac{\omega^2}{2g} = h_c + \frac{K^2{}_\omega h}{2g}.$$

Whence

$$h_c = \left(1 - \frac{K^2{}_\omega}{2g}\right) h$$

and

$$\frac{h_c}{h_k} = \frac{\left(\frac{1 - K_\omega{}^2}{2g}\right) h}{\left(\frac{K_\omega{}^2}{2g}\right) h} = \left(\frac{2g}{K_\omega{}^2} - 1\right).$$

From the table in the text the value of Kω corresponding to a given ratio $\frac{R_o}{R_1}$ is obtained to suit the selected ratio $\frac{c_1}{c_o}$;

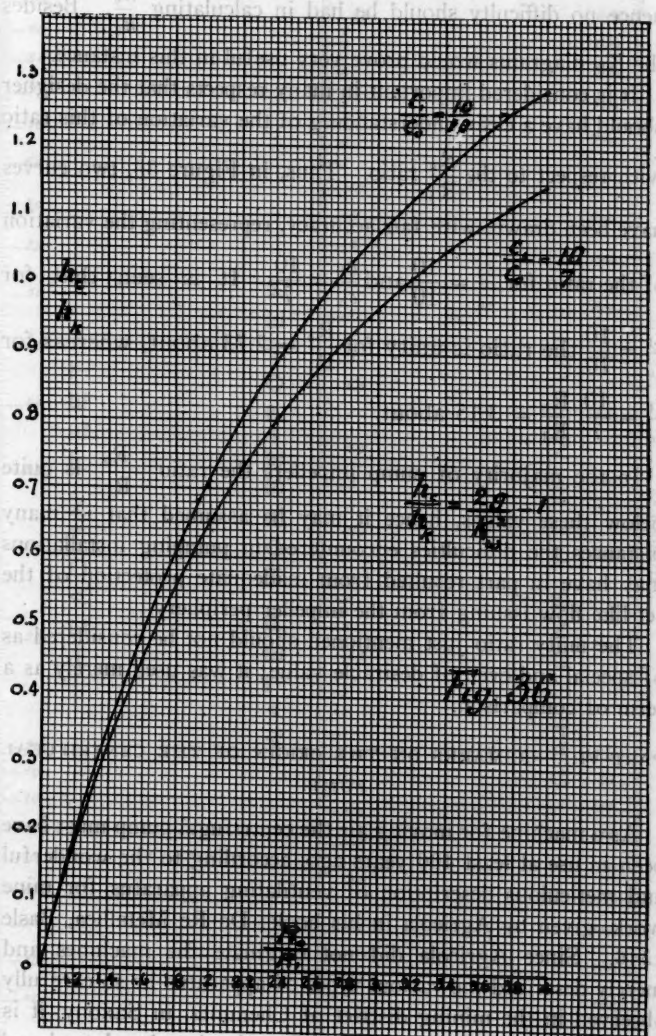

1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
$\frac{c_2}{c_0} = \frac{w_2}{w_1}$
$\frac{c_2}{c_0} = \frac{w_2}{w_1}$
$\frac{h_2}{h_1}$
$\frac{h_2}{h_1} = \frac{2g}{c_0^2} - 1$
$\frac{r_2}{r_0}$

Fig. 36

hence no difficulty should be had in calculating $\frac{h_o}{h_k}$. Besides the Kω diagrams would prove very useful in this instance.

It is considered important in many respects that the designer should have a clear understanding of the variation of this ratio with respect to the $\frac{R_o}{R_1}$ ratio. Thus, in Figure 36, two curves have been drawn, with fair accuracy, representing the variation of the ratio for $\frac{c_1}{c_o} = \frac{10}{10}$ and $\frac{c_1}{c_o} = \frac{10}{7}$. It is seen that for $\frac{c_1}{c_o} = \frac{10}{10}$ the ratio is unity for $\frac{R_o}{R_1} = 2.73$ about, whereas for $\frac{c_1}{c_o} = \frac{10}{7}$, $\frac{R_o}{R_1} = 3.15$ about.

In the majority of pump impellers the ratio $\frac{R_o}{R_1}$ is quite below these values, hence it may be assumed that in many instances the difficulties encountered in pumping installations may have in part resulted from inadequate protection of the jet-like flow issuing from the impeller periphery.

The unity ratio just mentioned should not be considered as of special and even of desirable value; it was used merely as a term of comparison.

PART VIII.—REMARKS ON THE ORIGIN OF THE CENTRIFUGAL PUMP.

As a machine for moving air the centrifugal pump must have been in use at least 400 years ago, according to the wonderful and meticulous description of ventilating apparatus for mine work, given by Agricola in his book: De Re Metallica, Basle 1546. Since Agricola did not originate the machines and merely described the various types of apparatus he so carefully observed in the mining district of Chemnitz, in Saxony, it is permissible to admit that they had been developed and employed

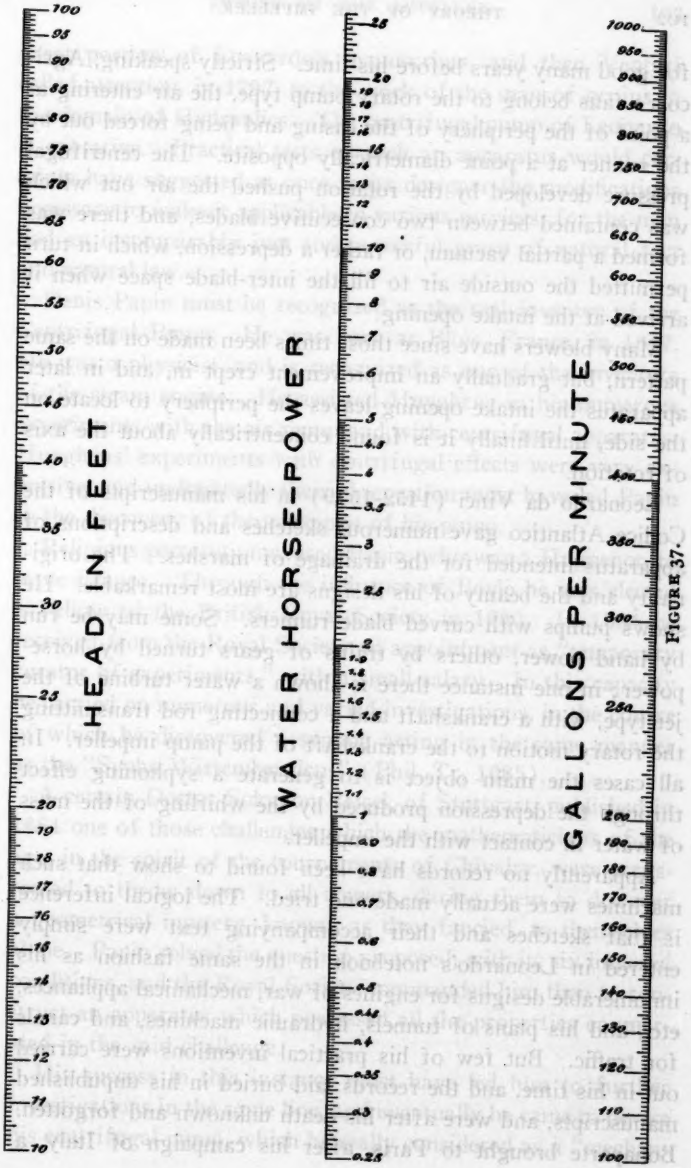


FIGURE 37.

for good many years before his time. Strictly speaking, Agricola's fans belong to the rotary pump type, the air entering at a point of the periphery of the casing and being forced out by the runner at a point diametrically opposite. The centrifugal pressure developed by the rotation pushed the air out which was contained between two consecutive blades, and there was formed a partial vacuum, or rather a depression, which in turn permitted the outside air to fill the inter-blade space when it arrived at the intake opening.

Many blowers have since those times been made on the same pattern, but gradually an improvement crept in, and in later apparatus the intake opening leaves the periphery to locate on the side, until finally it is found concentrically about the axis of rotation.

Leonardo da Vinci (1452-1519) in his manuscripts of the Codice Atlantico gave numerous sketches and descriptions of apparatus intended for the drainage of marshes. The originality and the beauty of his designs are most remarkable. He shows pumps with curved blade runners. Some may be run by hand power, others by trains of gears turned by horsepower; in one instance there is shown a water turbine of the jet type, with a crankshaft and a connecting rod transmitting the rotary motion to the crankshaft of the pump impeller. In all cases the main object is to generate a syphoning effect through the depression produced by the whirling of the mass of water in contact with the impeller.

Apparently no records have been found to show that such machines were actually made and tried. The logical inference is that sketches and their accompanying text were simply entered in Leonardo's notebook in the same fashion as his innumerable designs for engines of war, mechanical appliances, etc., and his plans of tunnels, hydraulic machines, and canals for traffic. But few of his practical inventions were carried out in his time, and the records laid buried in his unpublished manuscripts, and were after his death unknown and forgotten. Bonaparte brought to Paris, after his campaign of Italy, a

great portion of Leonardo's manuscripts, and then Venturi called attention, in 1797, to the work of the man of genius in the domain of Hydraulics. The centrifugal pump of Leonardo is operative. Practical tests of such an apparatus would certainly have suggested at once to its designer the modifications necessary to make it applicable to various services, for the man had an incomparably just and powerful grasp of natural fact and natural law.

Denis Papin must be recognized as the real inventor of the Centrifugal Pump. He was born at Blois, France, in 1647. He was a physicist, and is recognized as one of the inventors of the steam engine. He assisted Huyghens in his numerous experiments with the air pump and with centrifugal apparatus. Huyghens' experiments with centrifugal effects were very extensive, and undoubtedly their observation must have led Papin to the discovery of the principles of his pump.

Religious persecution caused Papin, who was a Huguenot, to leave France. Through the influence of Boyle he was elected a Fellow of the British Royal Society in 1680. In 1684 he received from the Royal Society an appointment as "temporary curator of experiments," with a small salary. In this capacity he carried on numerous and varied investigations, in the course of which he discovered a syphon acting in the same manner as the "Sypho Wirtembergicus" (Phil. Tr. 1685).

A certain Doctor Solomon Reisel, of Stuttgart, published in 1684 one of those challenges which the mathematicians of the age, in the spirit of the tournaments of Chivalry, were accustomed to throw down to all comers, daring them to discover a geometrical mystery, known, as they fancied, to themselves alone. Papin solved the question proposed, with its six imposed conditions, and the Royal Society commanded him then to construct an apparatus which possessed all the properties enumerated in the said challenge.

His success in this instance must have led him to further investigations in the same line, and eventually he came to devise his centrifugal pump, which he really considered as a "mechan-

ical syphon." It seems that he wanted to devise a positive syphon, simple in construction and possessing more marvelous qualities than the apparatus of Reisel. He then made a direct and beautiful application to a syphon tube of the centrifugal effects he studied with Huyghens. A complete description of Papin's invention is given in "Acta Eruditorum," Leipsic, June, 1689; but unfortunately this description is in Latin, and it is surmised that on that account it quickly passed into oblivion, since each of the subsequent inventors practically attempted to "start all over again," being apparently in complete ignorance of Papin's invention. The publication of the description under the auspices of the Royal Society, being a recognition of the invention, amounted practically to the granting of Letters Patent. A translation of part of the description and of a subsequent paper (1705) is given further on. This will end our historical sketch as regards the apparatus. For more extended details the reader is referred to a paper on the "Genesis of the Centrifugal Pump" prepared by the writer, and read by him at a meeting of the Colorado Scientific Society, April 3, 1915. (Proceedings, Vol. XI, pp. 49-102.)

ACTA ERUDITORUM,

Leipsic, June, 1689.

DESCRIPTION AND DEMONSTRATION BEFORE THE PRINCELY COURT OF CASSEL OF DENIS PAPIN'S LIFTING AND FORCING ROTOR.

The machine consists of one shallow cylinder AAAA, Figures 38 and 39, through the center of which passes the shaft BB. To this shaft are fixed the blades CCC, extending from the axis to the inner circumference, and in such a way that with the casing fixed, the blades can freely rotate. It is obvious that the said shaft can rotate in either direction; however, it is necessary to rotate the water contained in the casing AAAA so as to make it recede continuously from the axis. It is true from the first principles of Philosophy observed by

Descartes, and which are Laws of Nature, *that a body put into motion and left to itself will tend to travel along a straight line; and also that in a circular movement the body tends to recede from the axis of rotation and to leave in a tangential direction.*

The casing being tightly closed, *with the exception of one axial opening through which the water is admitted from the outside* and another at the circumference of the casing, delivering into a tube AD fixed in a tangential direction, it follows, then, that the water rotating in the casing AAAA forces itself through the tube AD, and thence ascends to the altitude which a body moving in the same velocity and in the same direction can attain.

If the outlet velocity is 32 feet per second, the water itself, neglecting the resistance of the air, can rise vertically to an altitude of 16 feet. With a greater or lesser velocity the altitude will be greater or lesser, but always proportional to the square of the velocity. This can be determined by the method of Galileo, given in his Dialogue on Mechanics. It is easy to calculate the velocity required to raise the water to a given height. Since the velocity can be made to increase indefinitely, no one can define the limit of action of such a wonderful machine, the proper motive power being supplied and the resistance of the air being neglected.

This invention will prove very useful, not only for supplying the water to residences, where a few men will suffice to handle the apparatus, but also in connection with water displays to be started at any time and kept going any length of time. The apparatus is very simple, and all there is needed is to lead the discharge pipe as required, and at a very small expense, to a suitably elevated reservoir, from which the water is led to the spouts and various places of delivery.

A shallow cylinder, with a diameter of about one foot, a depth equal to one inch, and with vanes suitably disposed, will prove superior when compared to the fire engine in use at the present time. This latter apparatus consists of two cylinders,

into which close fitting plungers travel, and is provided with four valves. It is easily seen that when the latter are choked with dirt, which prevents them from closing tightly, the greater part of the power exerted is then lost.

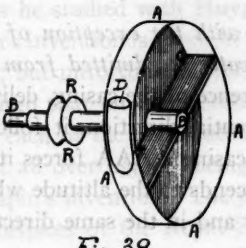


Fig. 38.

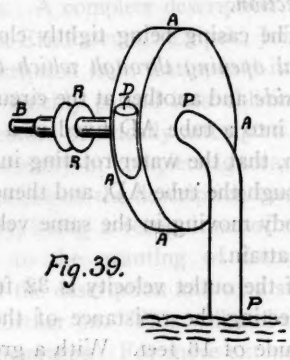


Fig. 39.

Fig. 40.

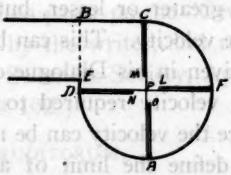


Fig. 41.

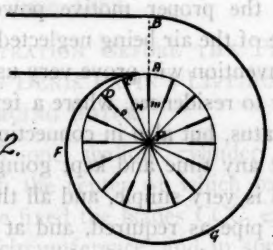


Fig. 42.

The new machine is powerful, and its cost moderate; it can be used in gardens for supplying hot or cold water; it can also be used in connection with horologes, it being adjusted so as to dip into the water, the latter by its own weight entering the axial opening, without suction.

In some instances it may be required to install a machine above the level of the water; this is shown in Figure 39, where the vessel AAAA, tightly closed, has its inlet pipe PP arranged so that its lower end is immerged into the water below; the machine itself, while in motion, causes the water to ascend into the pipe PP, to replace that delivered through the discharge pipe AD.

Difficulty may be experienced in sealing the side of casing AAAA, through which the shaft BB passes; it is necessary to prevent the air from passing through the cavity in which the said shaft rotates. Were the air to enter, the water could not ascend through the pipe PP. It will be easy, however, to surmount the difficulty by employing leather which, formed in the shape of a ring, will press against the shaft and the casing and thus exactly fill and seal the space. In order to more completely insure the sealing, a number of such rings can be used, the space between them being filled with grease or other matter just as thick, which would remain there always. This place must be, more than any other in the machine, provided against the ingress of air. The other parts being securely closed and the shaft put in motion, the water will ascend through pipe PP and thence, pressed by the vanes it will be forced through the pipe AD. This machine, performing like that of Reisel, should be called "Lifting and Forcing Rotor."

I have omitted here to show the framework onto which the apparatus is installed, and how the rotary motion is produced, but, according to circumstances, various ways may be chosen from. Thus my first model was not provided with a base, it being a crude little machine, held in place when to be used, with a cord passed around the pulley RR fixed firmly on the shaft. Then in this little machine the pulley caused the shafts and vanes to rotate, and it was observed that the faster the motion communicated, the greater was the force with which the water issued from the tube DD.

Besides the uses stated above, it may be added that this machine can work very much better than bellows, when actioned

by river current, weights and other force, all of which can more easily be applied to it than to the old style bellows.

It does not require such an alternating movement as is necessary to open and close the bellows; neither is it necessary to use leather, which is easily deteriorated by heat. When it is required to displace air, whose weight is always very small, a small power is sufficient to communicate a very rapid motion to the machine, and thereby produce a flow of very great intensity. The Prince was soon able to perceive the great variety of work that this new invention can accomplish. Since so much water can be delivered through the discharge pipe, a corresponding quantity of air can in the same way be handled.

If proper precautions are observed for the inlet pipe, so that the ingress of water be free from obstructions, it is obvious that the delivery pipe should be designed with the same care, and thus this machine will be found suitable for many other kinds of work besides those enumerated.

I believe nobody can point to an apparatus simpler and less liable to dearrangement than this one, and therefore when our kind Prince recognized my claims, his action afforded me the greatest pleasure.

I have observed that the water can be raised to a greater level by causing it to flow through the discharge pipe than it would reach were it allowed to issue as a jet. The delivery pipe should be so proportioned that as the water ascends with diminishing velocity, the cross-sectional areas should so increase that for each unit of time the same quantity of water would flow through any given cross-section (Figure 40).

A mold can be made so that a nozzle of this form be cast in metal. It may be preferable, in case the cast nozzle should weigh too much, to construct it of sheet metal properly assembled and so formed that cross-sectional areas be proportioned as described. It must be noted that when the cross-sections figured are not placed normally to the axis, the tube becomes distorted and is not proper to suit the requirements. For that reason it is not necessary to form the tube the whole length to

which the water can reach perpendicularly; a straight pipe may be used to a certain height, and thence from terminate in flaring out as per curve DXY, etc. It may also prove to be costly, both in time and labor to make all the tubes as per the correct method, and principally so when the water is to be used at different altitudes and through different pipes; then common pipes can satisfactorily be used. However, the full benefit of the remarkable properties just exposed can only be obtained by the use of the proper discharging cone. The weight of the ascending column of water is always overcome by the momentum communicated to the water, so long as there is no lateral issue in the column.

In other machines for raising water to great heights much difficulty is experienced in making the pipes, which require to be constructed of very strong material; otherwise the weight of the water flowing continuously through, would, in course of time, expand and distort them.

The shape of the delivery nozzle can be obtained by having a vertical jet of water issuing between a source of light and a perpendicular plane, so that the outline of the jet be projected on the plane, or screen, the image thus obtained would be very little out of true.

An instrument could be constructed which would delineate the desired curve DXYZ for small and large sizes of nozzles, the diameter and length of pipe being divided into the same number of equal parts, as, for instance, the diameter into 16 inches and the length into 18 feet.

I am preparing a number of experiments to be tried with my machine, and should anything new occur during the tests I shall duly present the facts to the judgment of the public. Meanwhile, I believe it will be seen whether, as Dr. Reisel declared, the present scheme is the same as that of the Wurtemberg syphon, or whether a difference can be found between the latter and the "Lifting and Forcing Rotor"; beyond this I keep silent. Of the two inventions, that which is capable of the greater number of applications should be publicly described.

COMMENTS.

Denis Papin reduced his invention to practice, for he constructed a working model which, when put to test, gave such good results that the inventor became very enthusiastic and, as is usual in such instances, predicted for the novel apparatus an assured supremacy over all other water or air moving machines. But he was not then able to run his pump impeller at speeds sufficiently high to permit of the detection of the inherent defects of his design and of the pump casing.

From the start he adopted, as the underlying principle of his apparatus the law observed by Descartes, viz: *in a circular movement a body tends (a) to recede from the axis of rotation, and (b) to leave in a tangential direction.*

In order to travel along a radial guide rotating uniformly about an axis, and to recede from this axis of rotation, *a body intended to traverse the casing and be delivered outside must necessarily enter the casing through an axial opening.* While such a statement may appear quite ludicrous and therefore unnecessary, because the matter is so self-evident, it is well to suspend judgment until the work of other earlier inventors is analyzed. Now, in view of what Agricola discloses in his book, it is clear that the older-types did not have an axial intake, and since in Papin's pump the inlet was axial, that in itself was an invention.

From the axial inlet the body, in this case water, was given, by contact with radial guides or vanes, a movement of rotation. This rotation should have and it surely did so, developed a centrifugal pressure in the casing, but that pressure was evidently neglected, because unexpected, by Papin. *His idea was that the rotation of the body caused the latter to recede from the axis and to travel radially along the guide until it reached the very end of the vane, when it would then leave in a tangential direction.* In order to guide the water as soon as it left the outer end of the vane, Papin provides, very properly, a tangential discharge pipe. This would have been very good

had the water always left the vanes at the same place in the casing, but it only accommodated each vane in succession. Papin thought that all he was getting out of his pump was, as we may say today, the kinetic energy of the water issuing from the vane tip in a tangential, hence straight, direction, and with a velocity equal to the peripheral velocity of the impeller. As a matter of fact, the mass of water made to rotate in the casing was under pressure through the outlet by virtue both of the centrifugal pressure and of the velocity of entrainment.

To make use of the high velocity possessed by the water issuing from the casing outlet, Papin simply invents the "diffuser." The discharge pipe, he tells us, is not complete unless it is made to terminate in some trumpet shape which can be very easily calculated and delineated.

It would perhaps seem idle to speculate as to whether Papin, with proper driving means of high speed and power at his disposition, could have realized the advantages of a discharge passage surrounding the impeller, receiving a continuous flow from the entire periphery of the latter, and so shaped as to facilitate the transformation of the kinetic energy into potential. While he might have actually so modified his original construction as to conform with the results of his observation, he would not have been able, in view of the status of science then, to explain correctly the action of his pump. However, in regards to this matter it behooves us to be very modest, since the centrifugal pump of today is very far from being correctly understood, even by its most reputed makers.

Quite worthy of remark, and entirely up-to-date, are Papin's advices concerning packing for the shaft and the care required to make the suction line airtight.. It is not uncommon to find the user of a pumping outfit claiming failure to operate, although the pump, when tested at the maker's shop, met the specified conditions. Diligent search by all concerned invariably brings to light the fact that air leaked through the line, either at some flange connection or through some crevice in the piping itself.

The arguments advanced by Papin against the plunger type of pump are the very ones in use today by enterprising pump agents, but as the world has grown slightly older and the centrifugal pump has obtained a good place of its own the arguments are now very much more aggressive.

While the first description of Papin's pump is of great interest, one is nevertheless filled with regret because this most unhappy man had not the necessary means to complete his invention. But if one looks further through the Transactions of the Royal Society, and arrives at the year 1705, another description is found, given by Papin himself, of a great improvement to his pump, which he then uses for handling air in mines. The apparatus is then complete, the casing is of the volute form, and only one thing is needed to make it entirely modern—that is, curved vanes instead of the radial ones he uses.

SECOND DESCRIPTION BY DENIS PAPIN (1705).

I am at present busy for a coal mine which has been left off because of the impurity of the air. I have therefore improved the Hessian Bellows, an account of which was printed at Leipzic, in Acta Eruditorum, ano 1689, with the title "Rotatilis Suctor Et Pressor Hassiacus," which may be applied for wind as well as water. The shape of the tympanum was cylindrical, as represented in Figure 41, here DAFC is the circumference, CP, DP, AP, FP are the radii which bear the wings CM, DN, AO and FI; CE is the aperture through which the air be driven in the direction of the tangent CB; and it may be observed that when the engine is working, every wing from the end of the aperture, till it comes to the beginning of the same aperture, C, drives always the same air with the same swiftness and at the same distance from the center so that in going over all that circumference the air finds resistance by friction and gains nothing at all.

I therefore form the tympanum in a special shape, as in Figure 42, where the spiral contour is AFGB, and it is to be

observed that every wing in going around drives new air, because the air which is first in motion finds room to recede from the center towards the spiral circumference, and so gives room for new air to come to the wing, and when the wings come near to the aperture they drive their new air into the aperture without any friction; also the air which has been first driven and removed from the wing cannot lose its swiftness, because the wings which continually follow do continually drive new air, keeping that which is before always in the same velocity. This new shape of the Hessian Bellows affords also another advantage, because the air in going around follows the spiral line, which is nearer to the straight line than a circular circumference, and when the air comes to the aperture it gets into it without any loss of its substance; but in the cylindrical machine, Figure 41, this air goes around along the circular path, and when it comes to the aperture it is not driven exactly in the direction of the tangent, except in the beginning at C, and afterwards the impulsion is oblique, which is always increasing till the wind comes to the point A, and must occasion a great diminution of its strength. I believe therefore that this spiral form is a good improvement to this engine. I have made such bellows where the radius AP is but 10½ inches, the wing AM 2 inches broad and 9 inches high. Because the tympanum is also so high, or little more, the aperture AB was also 9 inches, or a little more, so that it makes a square hole. When I work this engine with my foot it makes such a wind as to raise up 2 pounds weight, and a stronger man could do much more, but this is more than sufficient for our purpose, since we need only drive air enough for the respiration of such men as can work in the mine; and we can easily make wooden pipes with boards to conduct the wind to the very bottom, so that the air within may be continually renewed as well as without.

I have made some trial of the Hessian Bellows in a very strong fire in a furnace to melt glass, iron or other hard metal, and though I could open the furnace above the matter to be wrought upon, yet no flame would get out through the aperture,

nor could cold air from without get into the furnace, so that it is very likely this will be a great convenience for several sorts of work; since men may work the matters when they are most softened in the fire; and they may be drawn up perpendicularly, so as not to be bent as they are when drawn horizontally. I believe this would be a good way to make glass pipes and looking glasses of extraordinary size, and I think that the Hessian Bellows may be applied to several good uses and so deserve very much to be improved.

ORIGIN OF THE THEORY.

Among the scientists, in the early part of the last century, who interested themselves to the problems presented by centrifugal machines, then quite rudimentary, Mr. Combes, an eminent French mining engineer, compelled by his very occupations to study means of ventilation in the mines, made public some important Memoirs which appeared in the "Annales des Mines," tomes XV and XVIII, in the year 1840. He considered that, apart from the physical properties of the fluids handled, the hydrodynamic phenomena of the motion of a fluid when traversing a rotating bladed drum, are alike for gases and liquids, and he established a mathematical theory which, aside from modifications of form in the equations, seems to have been adopted universally. This theory resolves itself to this, that since the bladed drum or impeller is the operative organ of the apparatus—pump or fan blower—that is, the energy producer, the amount of work it may furnish when operated at a given and constant speed equals the difference between the quantities of energy measured at the periphery and at the vane inlet. The centrifugal pressure generated on the mass of fluid in the rotating drum produces potential energy, and the entrainment of the fluid by the drum furnishes kinetic energy; the sum of these two represents the total work done, which may be readily expressed thus:

$$\text{Work} = \text{W H} = \text{W}\left(\frac{p_o - p_1}{d}\right) + m\left(\frac{u_o^2 - u_1^2}{2}\right).$$

W is the weight of fluid handled per unit of time;
m is its mass;
H is the height to which the mass is raised;
p_o, p_1 = pressure per unit of area at outlet and inlet, respectively;
d = specific weight of the fluid;
u_o, u_1 = absolute velocity at outlet and inlet, respectively.

Obviously the total head produced is obtained by eliminating W, thus:

$$H = \frac{p_o - p_1}{d} + \frac{u_o^2 - u_1^2}{2g}.$$

The value of $p_o - p_1$ is obtained by integrating the work of the centrifugal force acting radially from the inlet to the periphery, but in accordance with the notions obtaining at that period. Finally the general equation for the head becomes:

$$H = \frac{V_o^2 - V_1^2}{2g} - \frac{w_o^2 - w_1^2}{2g} + \frac{u_o^2 - u_1^2}{2g}.$$

V_o, V_1 = peripheral velocity at outlet and inlet;
w_o, w_1 = relative velocity at outlet and inlet.

Once the inlet and outlet angles are determined in function of the head and of the peripheral velocities the problem is usually considered as solved, because the delineation of the impeller is then a question of routine calculation and drawing, to facilitate which practically all authorities on the subject advise that the shape of the vanes is a mere matter of sentiment and taste. The angles can be obtained by ordinary trigonometrical process.

UNWIN'S THEORY.

In a paper "on the Centrifugal Pump" presented before the Institution of Civil Engineers, in 1877, Professor W. C. Unwin remarked that "* * * it is somewhat singular that no tolerably complete theory of the action of the pump exists in English engineering literature; and though theories on the centrifugal pump are to be found in some French and German

writings, these are generally restricted to the examination of some particular type of pump, and upon assumptions (made to simplify the reasoning) which are not of a very satisfactory character * * *."

Professor Unwin proceeds then to establish his theory of the pump in a simple manner, as follows: "In the action of the centrifugal pump disc on the water it is only the forces which produce rotation which do work, for from the symmetry of all parts of the disc the radial forces must be in equilibrium. Let any mass m of water flow through the pump disc in t seconds, and let its velocity of rotation be changed from V_1 to V_2 whilst it passes from the inner radius r_1 to the outer radius r_2. Let P be the force acting at radius r to rotate the pump disc. Then, from the equality of angular impulse and change of angular momentum,

$$P r t = m (V_2 r_2 - V_1 r_1).$$

Or, if Q is the volume flowing through the pump in one second, and G the weight of a cubic foot of water,

$$P r = \frac{G Q}{g}\left(V_2 r_2 - V_1 r_1\right).$$

If ω is the angular velocity of the disc, then the work expended by the disc on the water is

$$P r \omega = \frac{G Q}{g}\left(V_2 r_o - V_1 r_1\right)\omega.$$

This equation includes the work expended in overcoming any frictional resistance to the motion of the Q cubic feet of water considered. It does not, however, include the work expended on other portions of water which may surround the pump disc, or in axle friction or belt friction. It includes, therefore, the friction within the wheel passages, but not the disc friction of the wheel covers.

The work $Pr\omega$, assuming the efficiency equal unity, represents wH, and $GQ = w$, whence

$$P r\omega = w H = \frac{G Q}{g}\left(V_2 r_2 - V_1 r_1\right)\omega = \frac{w}{g}\left(V_2 r_2 - V_1 r_1\right)\omega,$$

and finally $H = \dfrac{(V_2 r_2 - V_1 r_1)\omega}{g}$.

On page 263 of Professor Unwin's paper his equation (13) gives the value $H = \dfrac{v_2 r_2 \omega}{g}$, which expressed with the notation symbols adopted here is

$$H = \frac{V_o[V_o - c_o \operatorname{cotang} \alpha_o]}{g}.$$

At the normal point where the water enters without shock, the relative component of the velocity is tangent to the first element of the vane and $c_1 \operatorname{cotang} \alpha_1 = 0$, hence $v_1 r_1 \omega = 0$.

In what particulars does this expression differ from the Combes' equation? It is easy to prove that there is no difference between the two, except one of form. Referring to Figure 13,

$$c_o^2 + c_o^2 \operatorname{cotg}^2 \alpha_o = u_o^2 \quad W_o^2 = c_o^2 + \omega_o^2$$
$$c_o \operatorname{cotg} \alpha_o = V_o - \omega_o \quad V_o(V_o - c_o \operatorname{cotg} \alpha_o) = V_o \omega_o$$
$$\therefore W_o^2 - u_o^2 = V_o^2 - 2\omega_o V_o \quad \therefore V_o \omega_o = \frac{V_o^2 - W_o^2 + u_o^2}{2}$$

and finally

$$\frac{V_o\,[V_o - c_o \operatorname{cotg} \alpha_o]}{g} = \frac{V_o^2 - W_o^2 + u_o^2}{2g}$$

Solving in the same manner for the conditions at inlet, the general equation is:

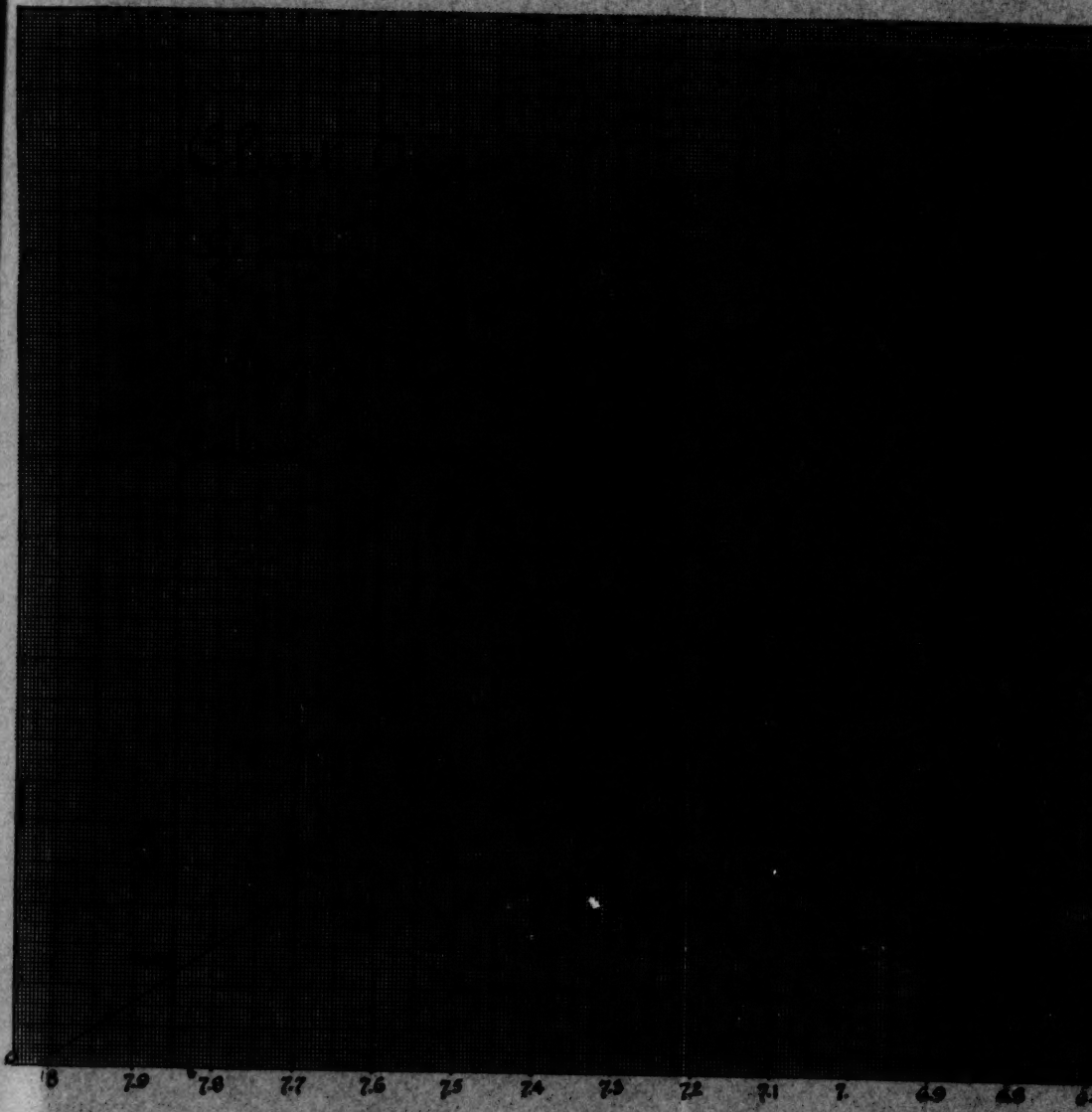
$$\frac{V_o[V_o - c_o \operatorname{cotg} \alpha_o] - V_1[V_1 - c_1 \operatorname{cotg} \alpha_1]}{g}$$
$$= \frac{V_o^2 - V_1^2}{2g} - \frac{W_o^2 - W_1^2}{2g} + \frac{u_o^2 - u_1^2}{2g}.$$

In view of this result Unwin's criticism of the older theories does not hold. It is curious to note that Unwin's formula, both in its original form and in some of its numerous trigonometrical

transformations have been sold several times to ambitious pump builders in this country, by devising individuals. In one instance, particularly, the representative of the buying concern returned to his employers after a period of study and observation of the shop method of the selling party, bringing with him the much desired theoretical information in the form of the XIIth volume of the Encyclopedia Britannicâ, in which the article on Hydrodynamics, by Unwin, contains the theory sketched above.

The same theory was evolved by quite a number of authors at various periods, for instance by J. Weisbach in 1851, Fink in 1878, etc. Professor Rateau brought it forth in 1891 and later in 1897, in his well known work on Turbo Machines, first published in the "Revue de Mécanique." On page 828 of that Revue (September, 1897) Professor Rateau states in part: "* * * I thought until now that the form given to this fundamental equation was novel when I published it in 1891. It seems, effectively, that it is not found in any work written in French before that time. But it is my duty to state that this form, so far as concerns centrifugal pumps and hydraulic turbines, was already known of the English authors, of which I was not aware."

It would be quite interesting to compare the Combes' formula with those developed in this work; anyone following this line of pump design may do it very readily and with fruitful results.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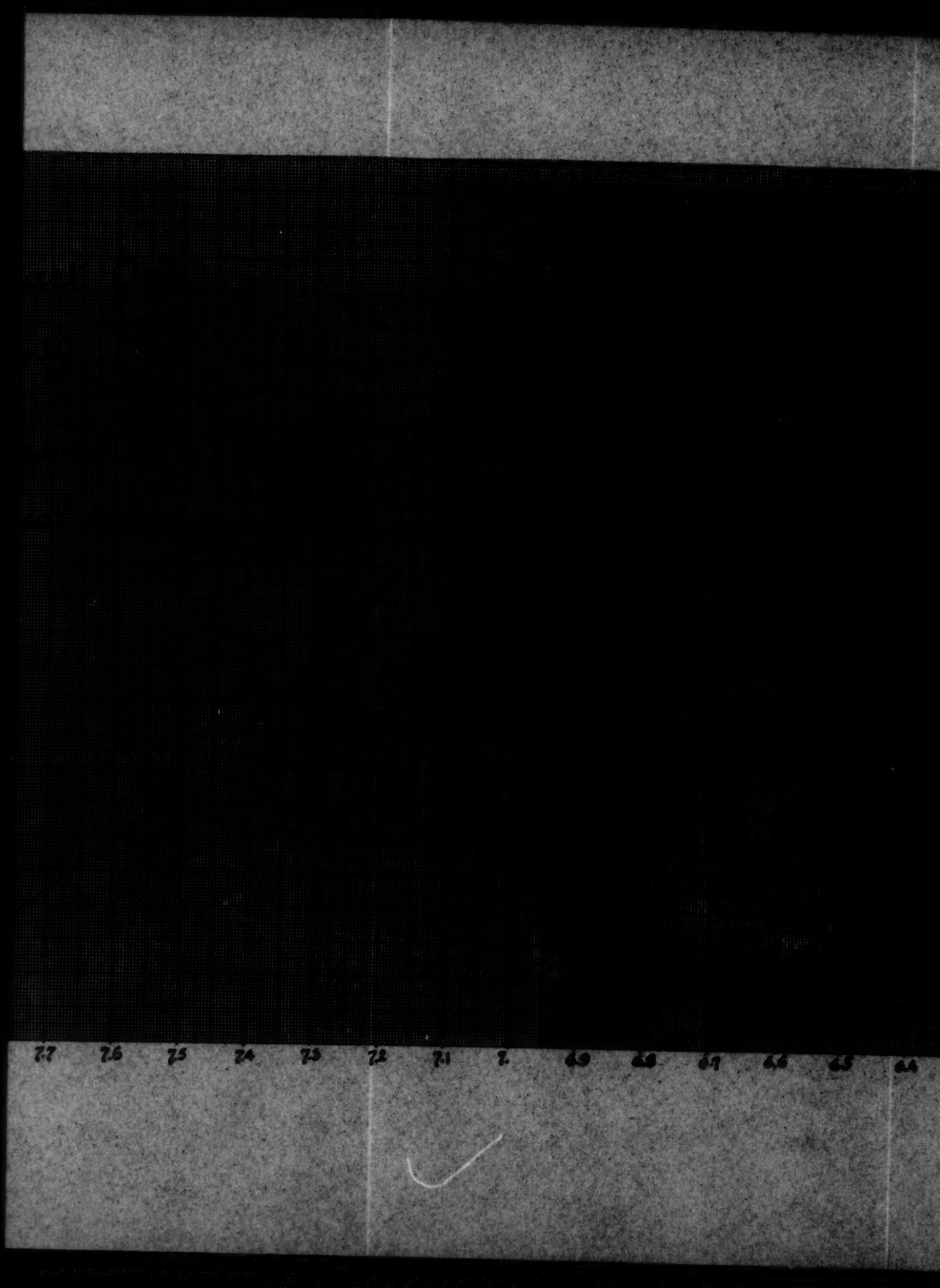
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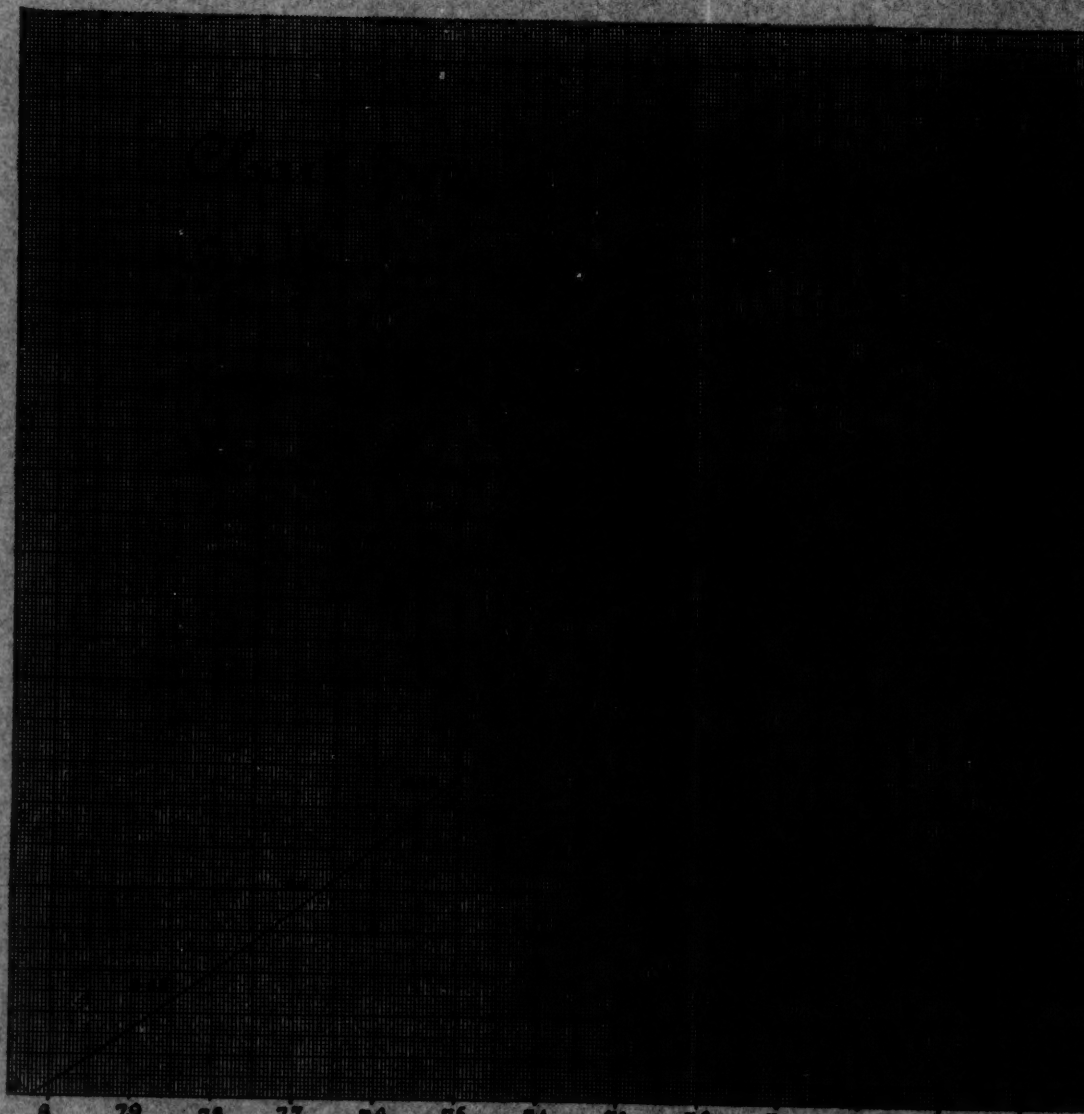
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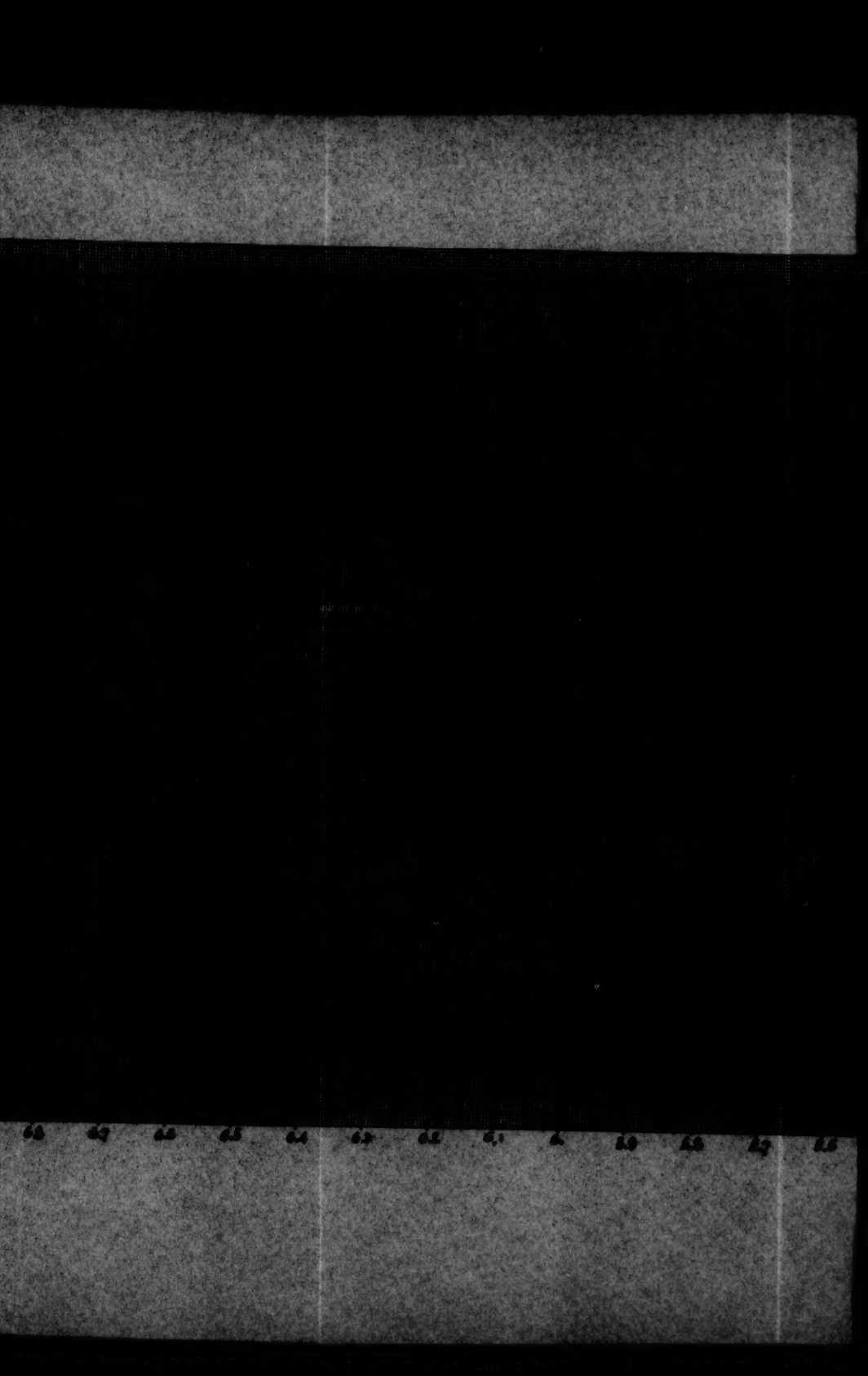

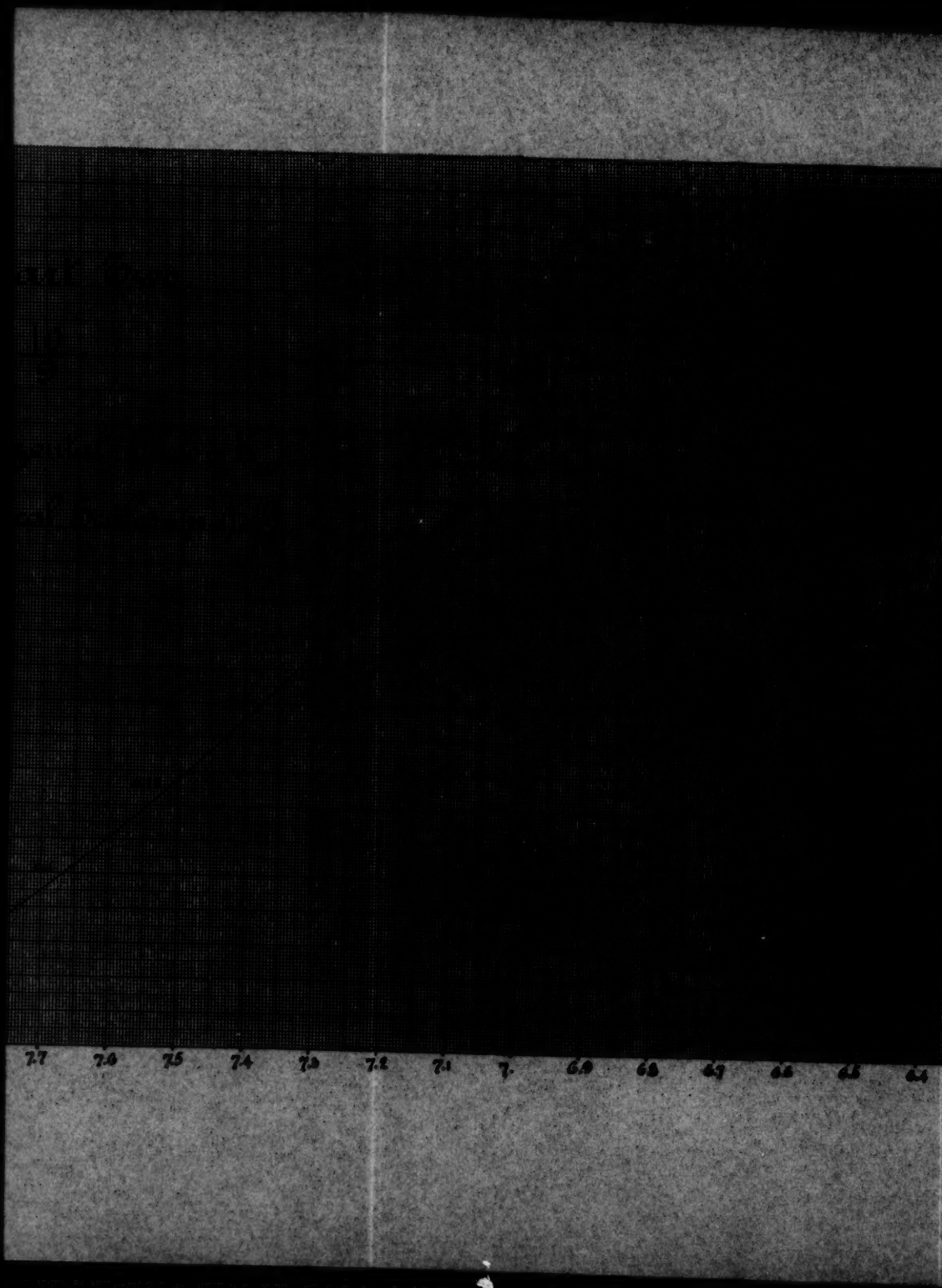
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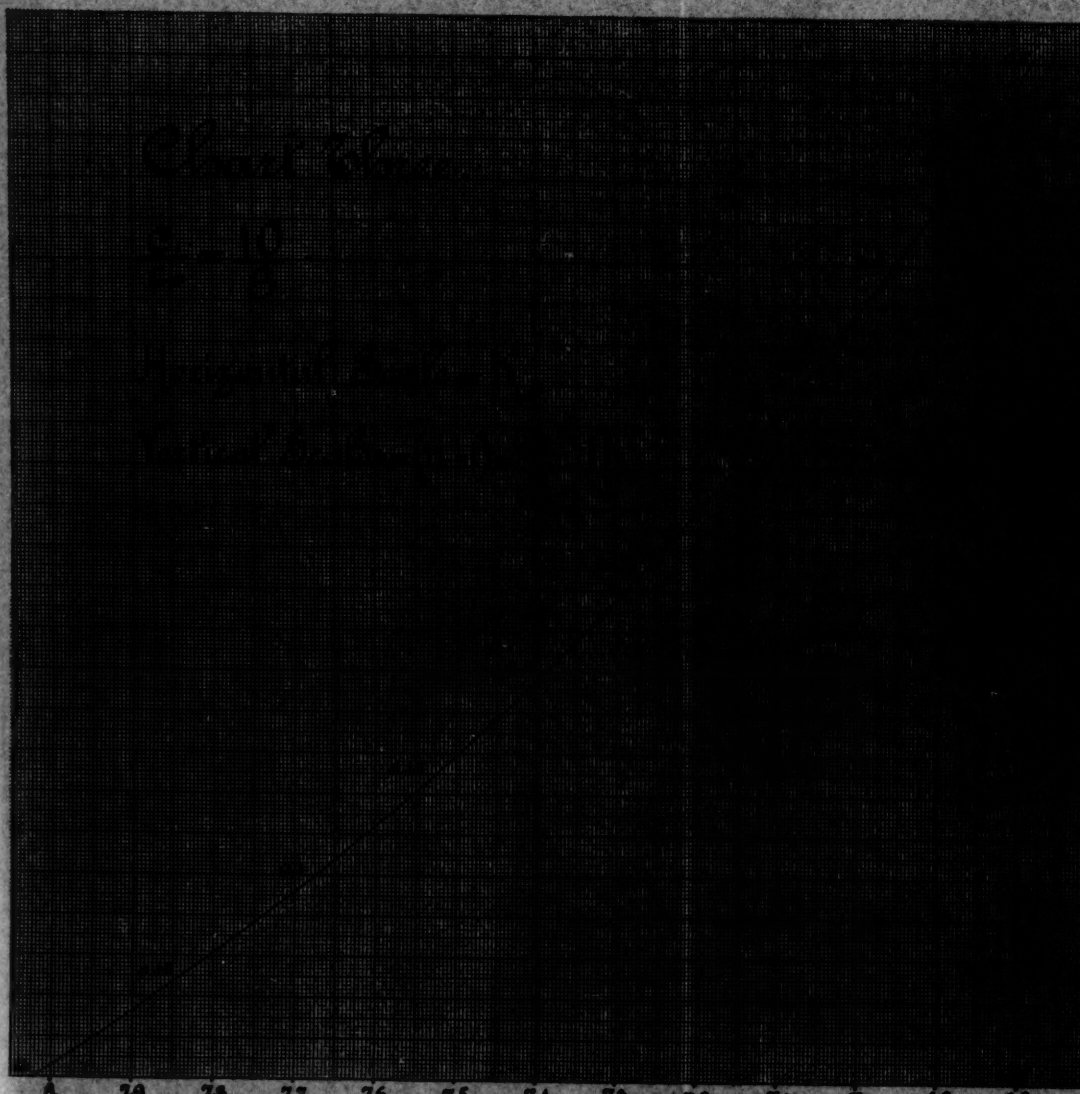
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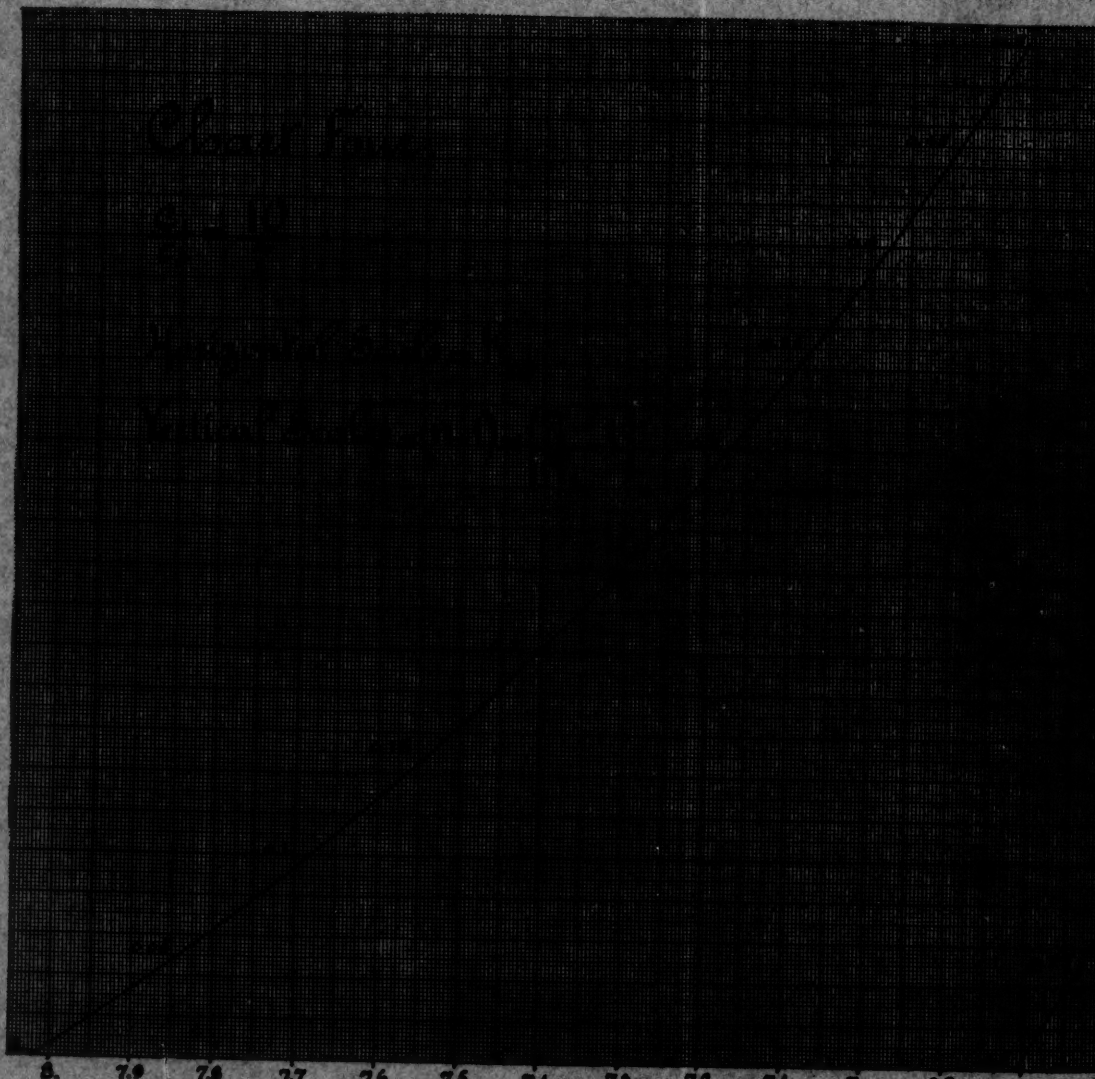
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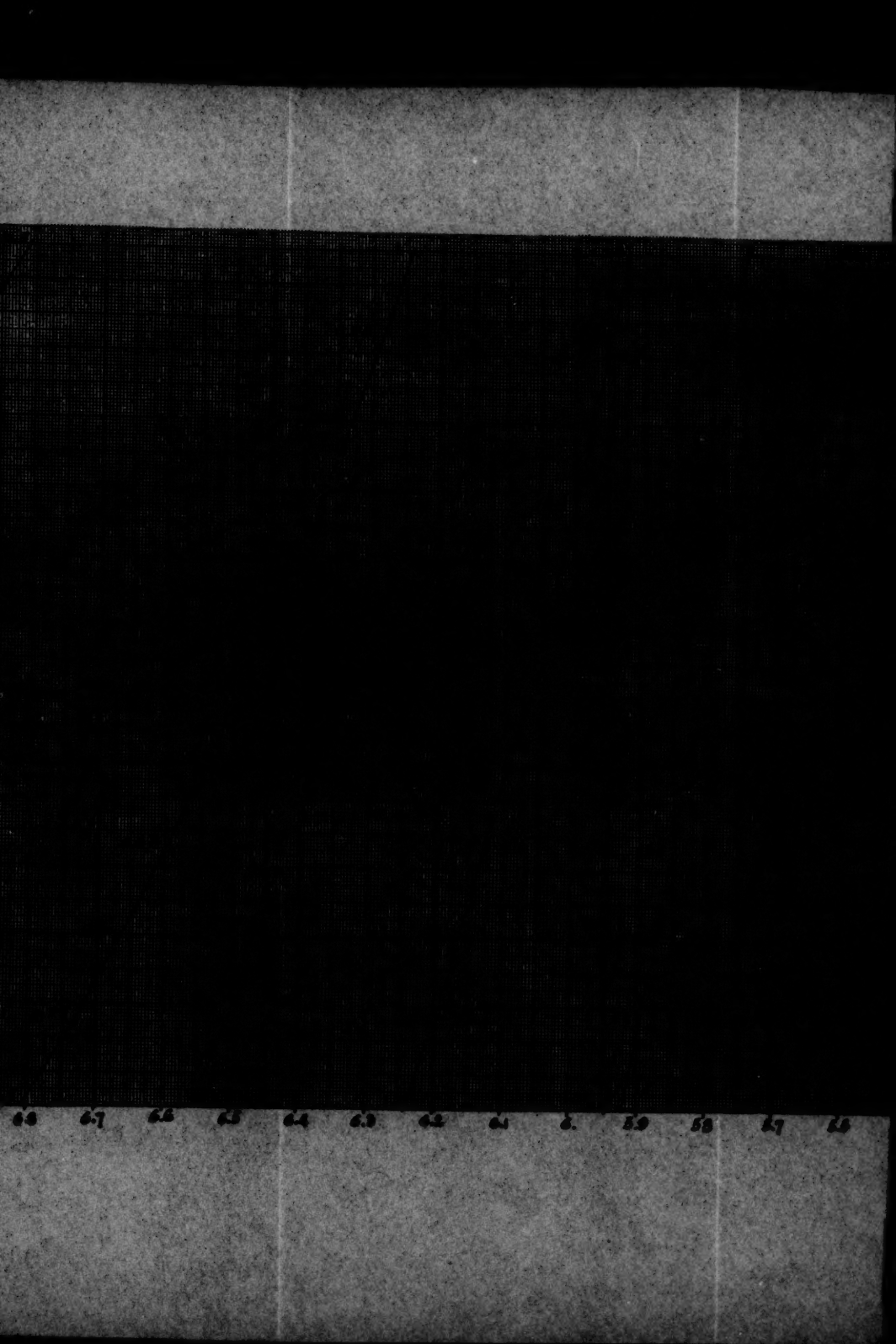

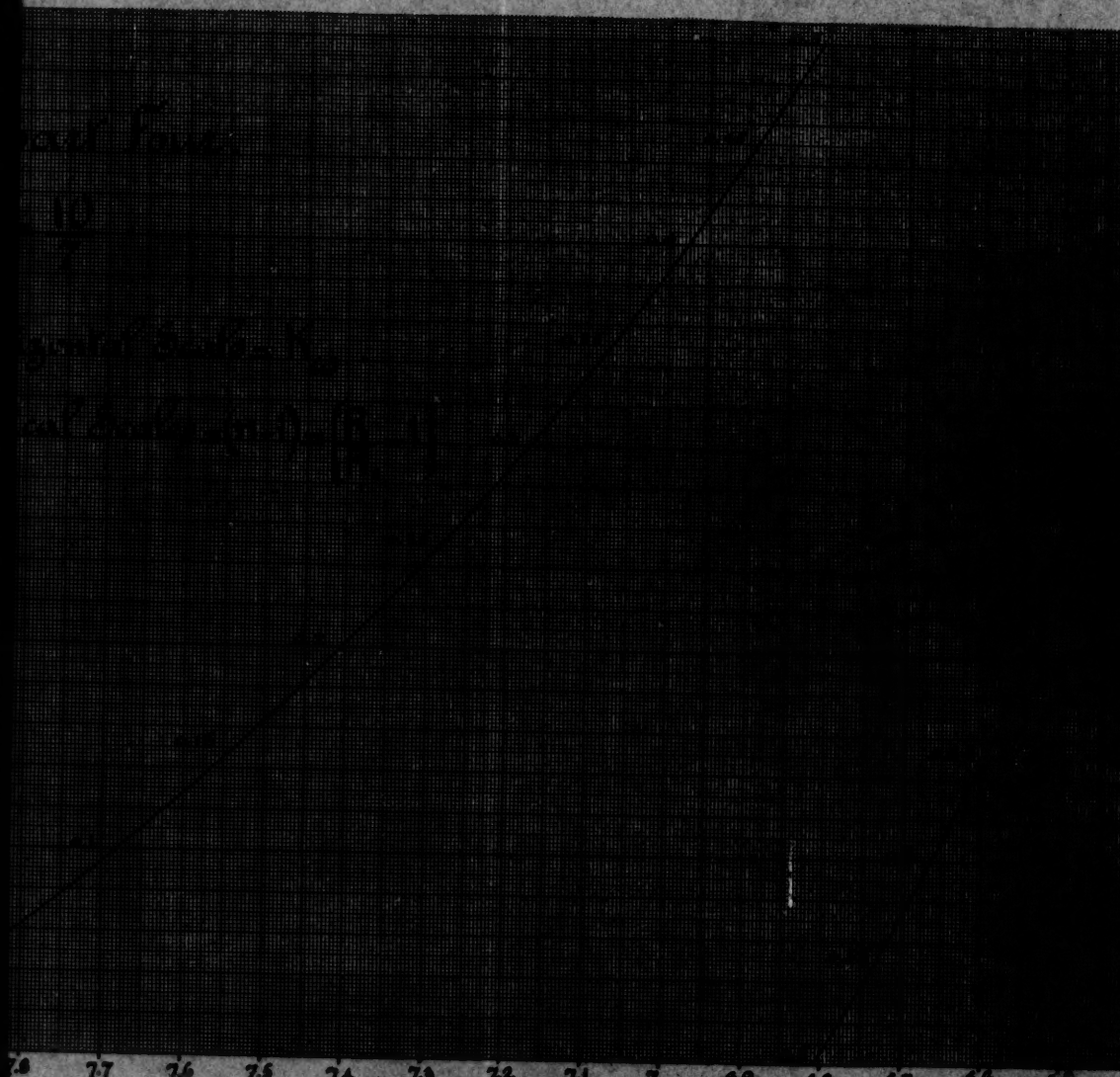

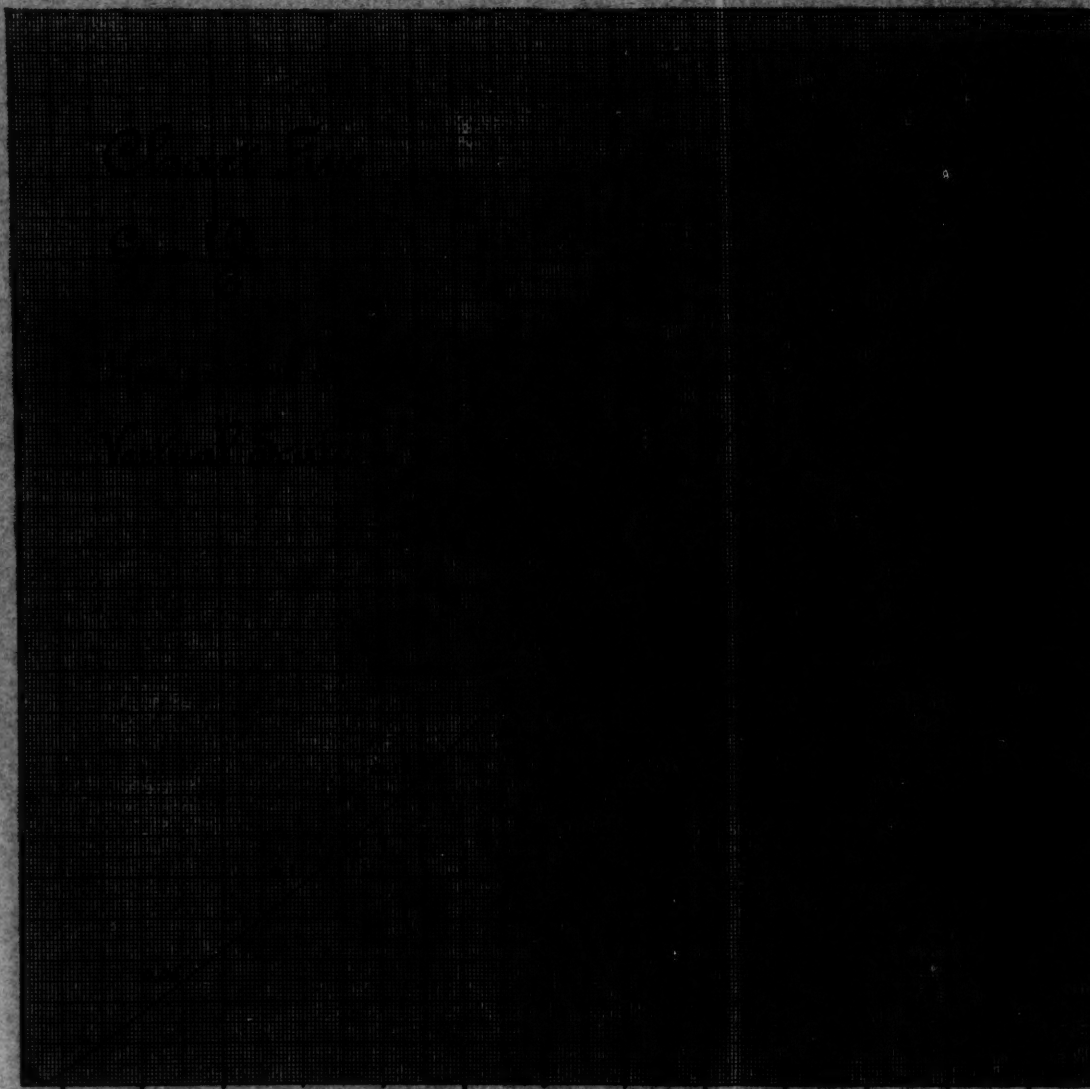
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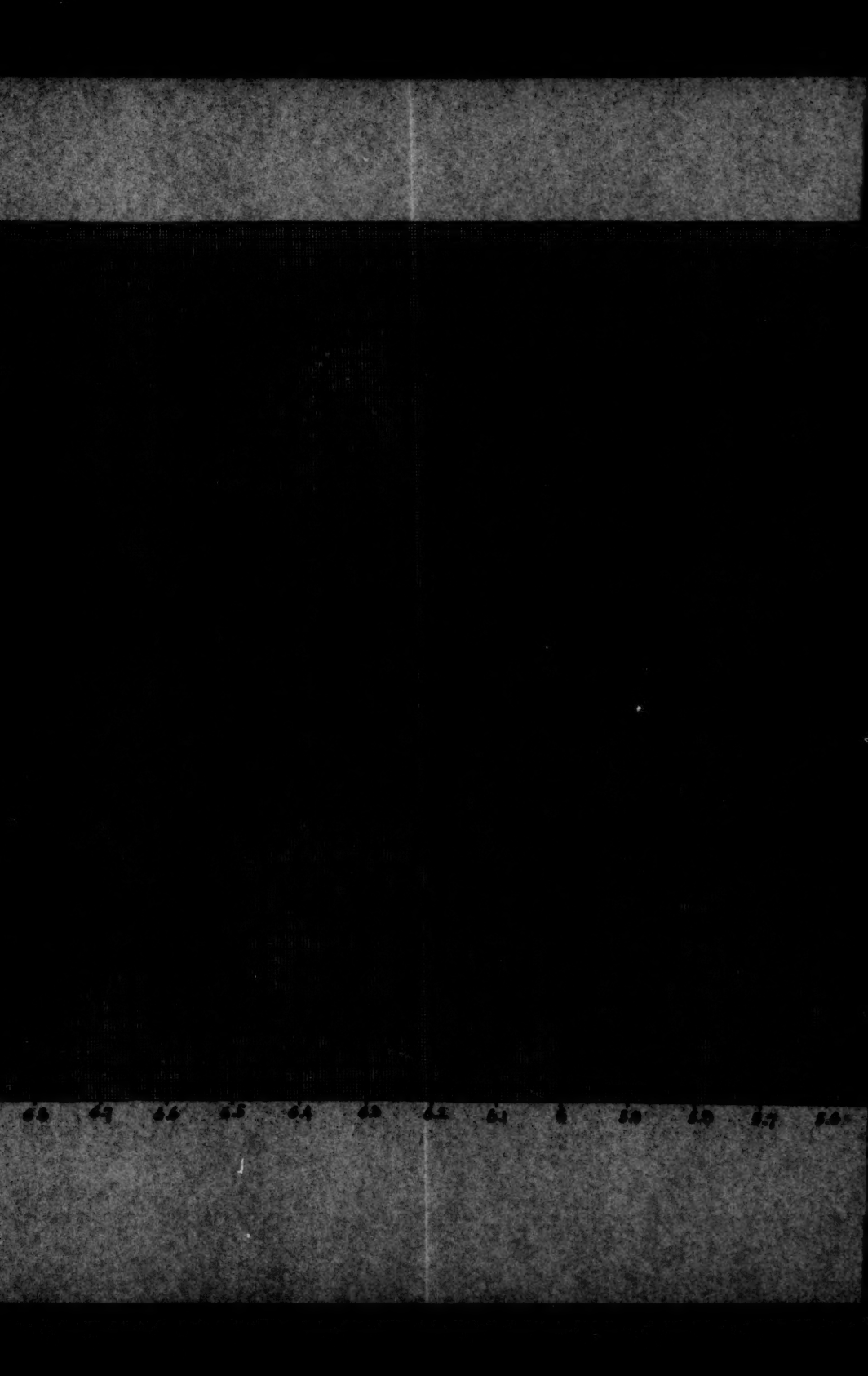

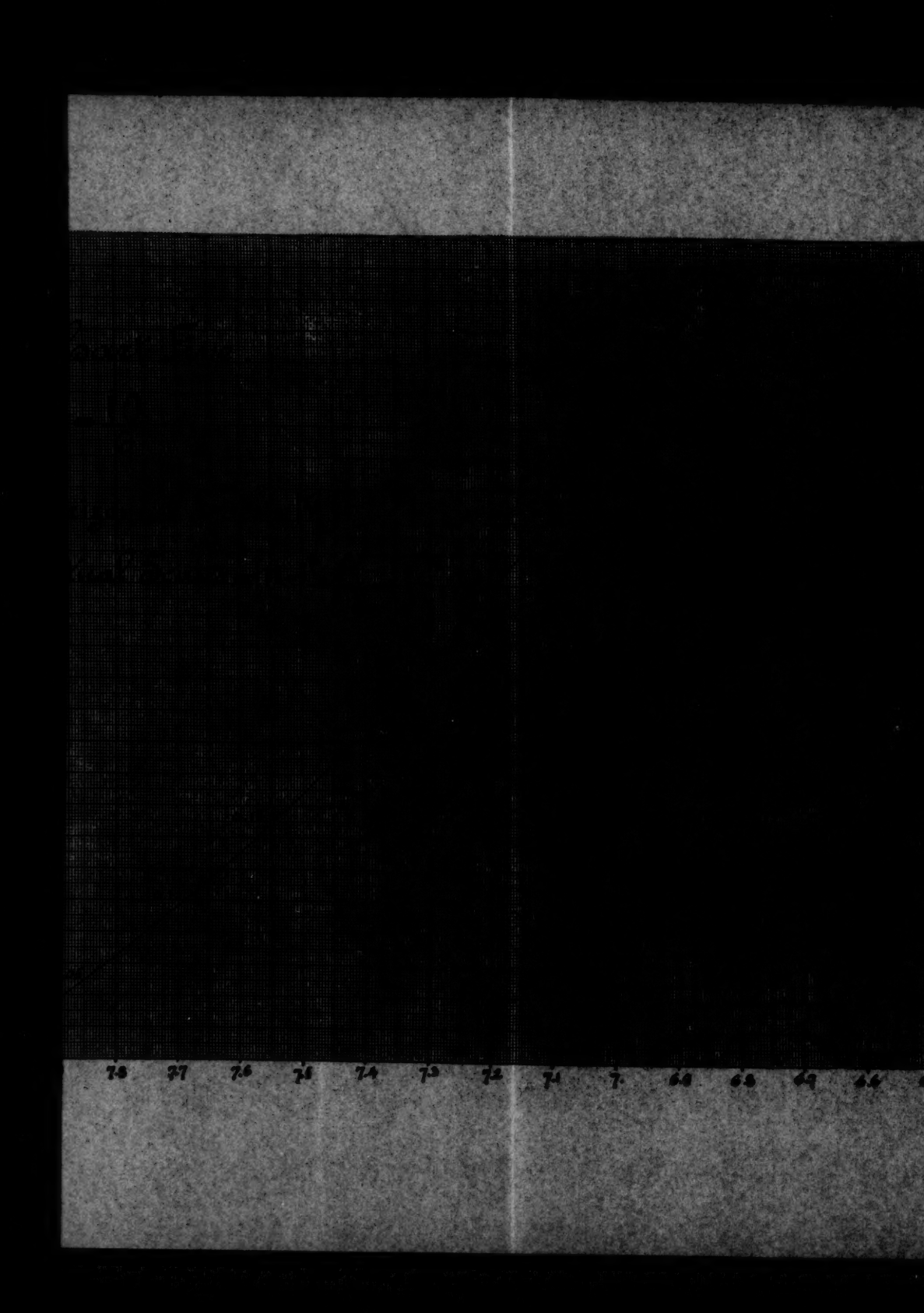

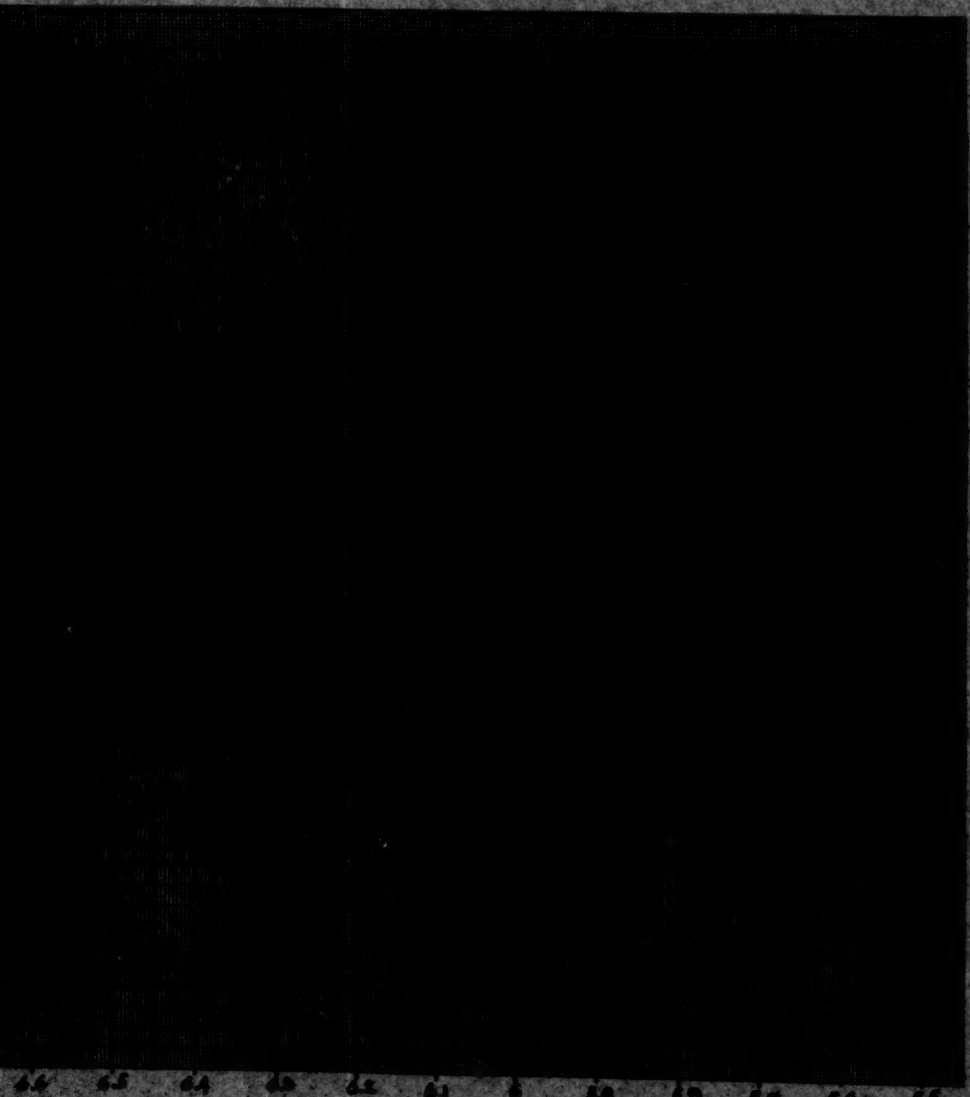

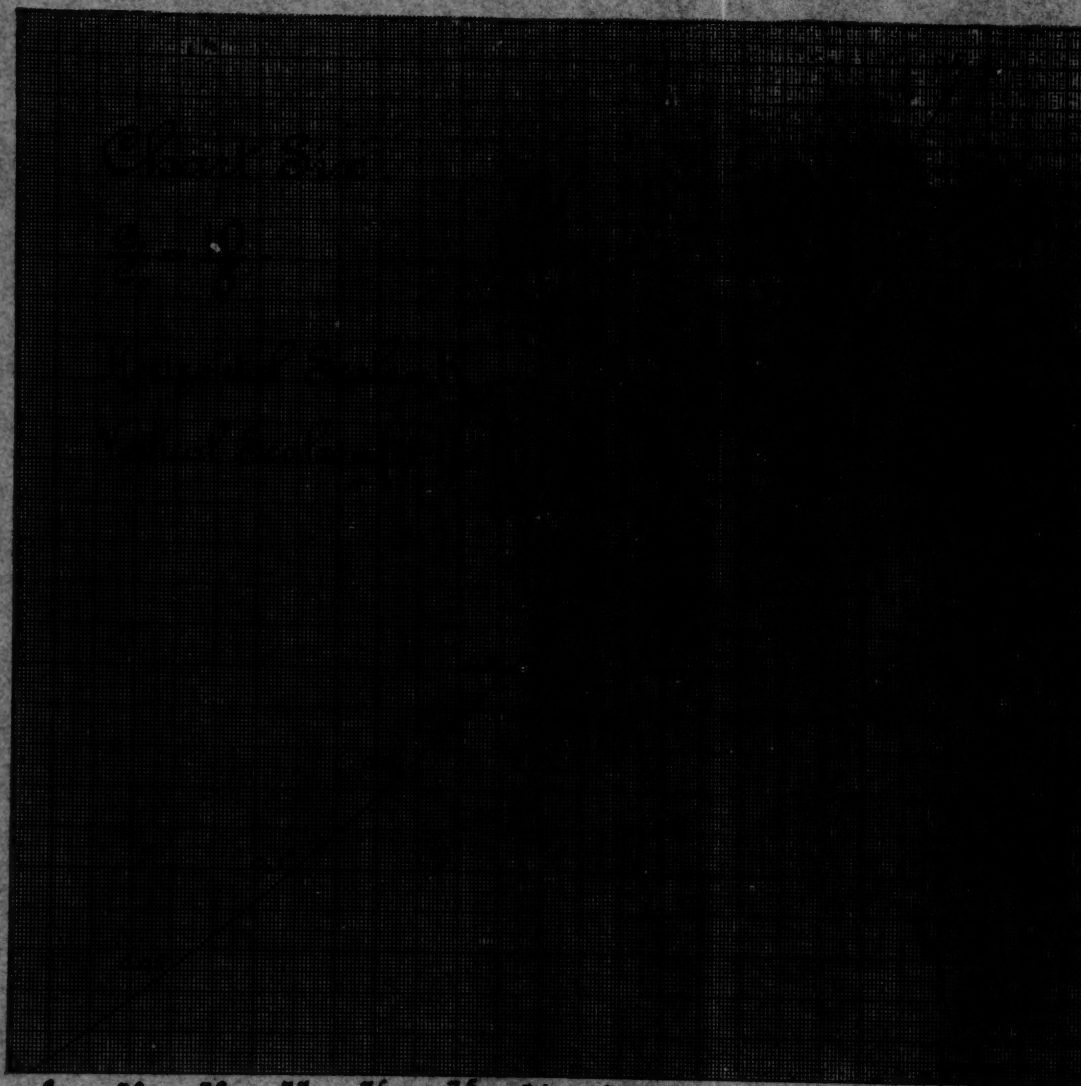
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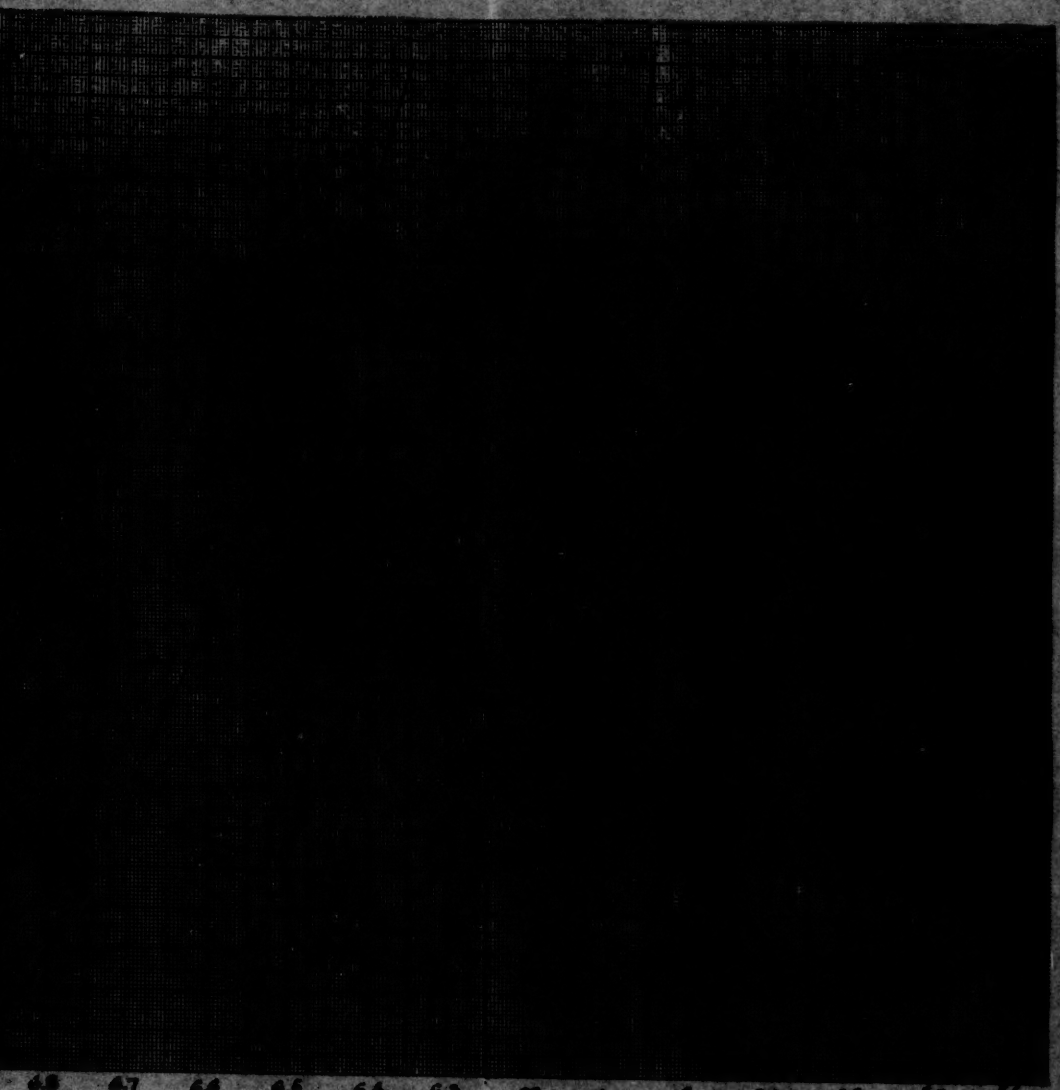


THE GRAPHIC RELATION OF SPECIFIC GRAVITY TO VISCOSITY IN THE CASE OF FUEL OILS.

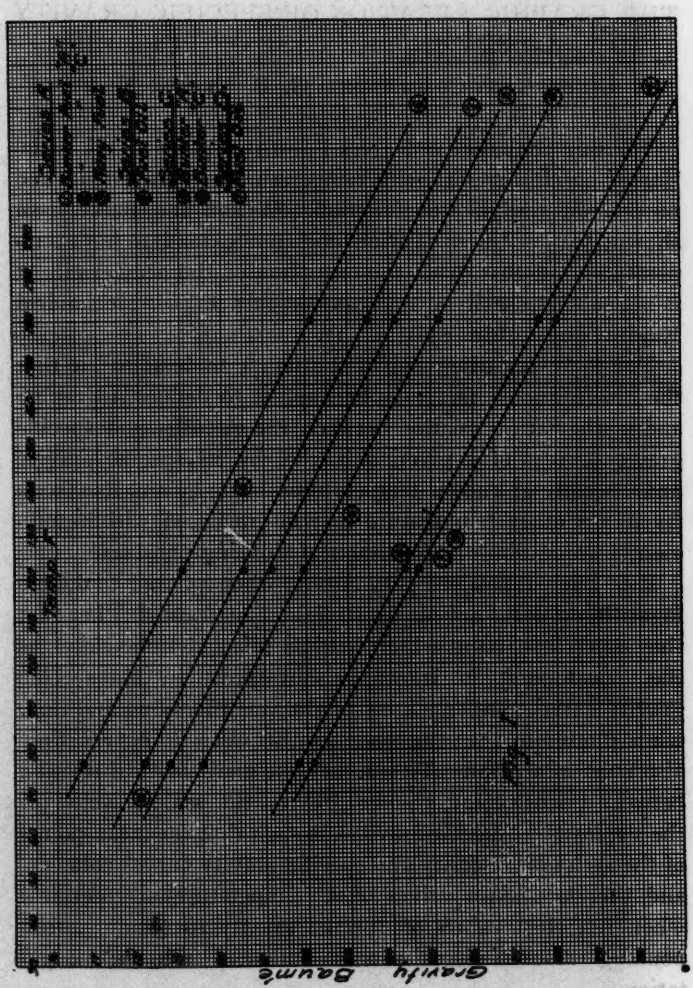
By Lieutenant-Commander G. B. Vroom, Member.

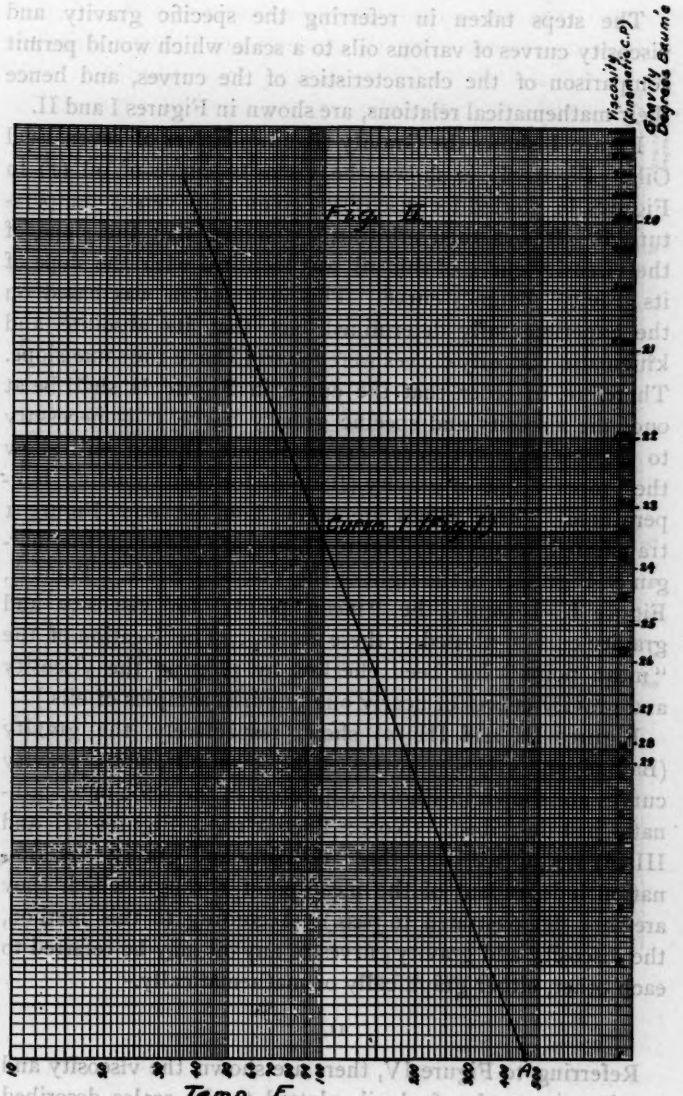
"Within the range 0 degrees F.—195 degrees F. the variation of specific gravity with temperature is to all intents and purposes a straight line function. Wilson and Balke have shown that the linear relation does not extend to high temperatures and that, with the exception of the heavier oils, it is not safe to use it beyond 200 degrees F."—"Mechanical Engineering," July, 1925, p. 537.

The following discussion will attempt to demonstrate that the above quoted fact may be applied to deduce a graphic and mathematical relation between the viscosity and specific gravity curves of a given oil; whereby, its specific gravity and viscosity at known temperatures being given, its viscosity at any other temperature may be predicted within sufficiently accurate limits for practical working purposes.

It is known that viscosity curves, when plotted to double logarithmic scale, appear as straight lines; and further that they meet in a point which will be referred to hereafter as the "mean point."* It can be shown, further, that specific gravity curves similarly are straight line functions and that, when plotted to the same scale as the viscosity curves, they also meet in a point. To simplify the mathematical relation between viscosity and specific gravity, the scale selected for the values of y (specific gravity), is such that the mean points of the series of viscosity and specific gravity curves coincide, and, therefore, since values of x (temperature) are identical when referring values of specific gravity to obtain corresponding values of viscosity, x is eliminated from the computation.

*Fuel Oil Viscosity—Journal of American Society of Naval Engineers, November, 1925.


Gravity Baumé


Viscosity (Kinematic c.p)
Gravity Degrees Baumé
Temp. F.

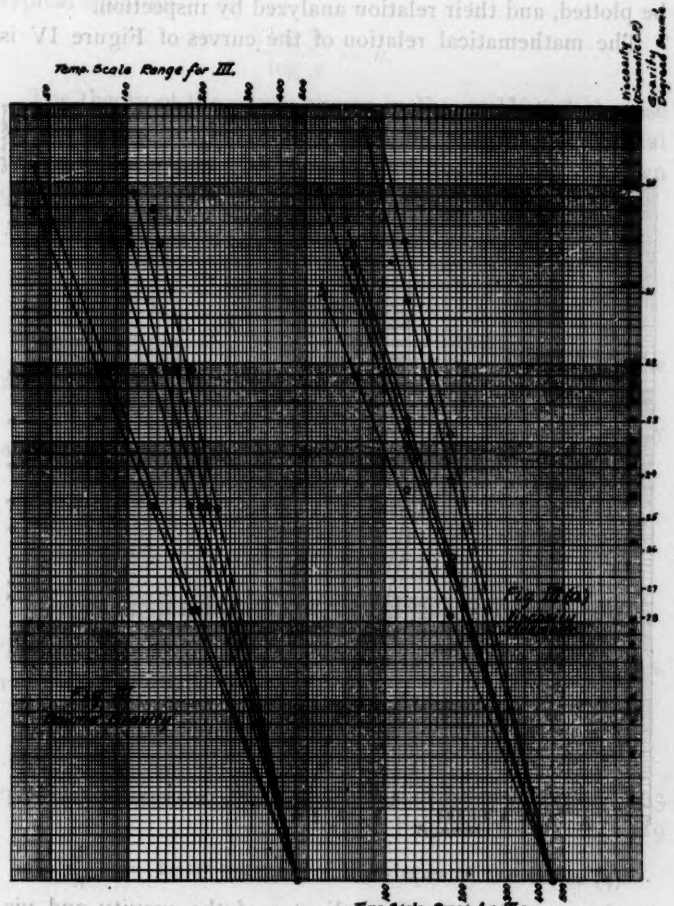
The steps taken in referring the specific gravity and viscosity curves of various oils to a scale which would permit comparison of the characteristics of the curves, and hence their mathematical relations, are shown in Figures I and II.

Figure I shows the Gravity (Baumé) curves of seven Fuel Oils. Figure II is shown the viscosity curve of oil (1) in Figures I: plotted for viscosity (kinematic) versus temperature F. It was then arbitrarily assumed that the slope of the viscosity curve of this oil was identical with the slope of its gravity (Baumé) curve. This assumption was based on the experimentally determined fact that its absolute and kinematic viscosity curves were found to have the same slope. Therefore, in order that the curve in Figure II shall be at once the viscosity and specific gravity curve, it is necessary to select a scale, for curve (1) in Figure II, that will show the proper values of gravity (Baumé) for corresponding temperatures. These values are taken from curve (1) Figure 1 transferred to curve (1) Figure II, and are shown in the margin of Figure II to the right of the Viscosity Scale. Point A, Figure II, becomes the "mean point" for viscosity and gravity curves, plotted to these scales. (The location of the "mean point" has been checked by plotting the viscosity and gravity curves of nearly one hundred samples of oils.)

Figures III and III (*a*) show, respectively, the gravity (Baumé) and viscosity curves of the seven oils, whose gravity curves are shown in Figure 1, plotted to the scale of co-ordinates just described. (Note: These curves, Figures III and III (*a*) are plotted from three viscosity and gravity determination independent of the mean point, through which they are thus demonstrated to pass.) According to this method, the viscosity and gravity curves of any oil may be referred to each other, either graphically or mathematically.

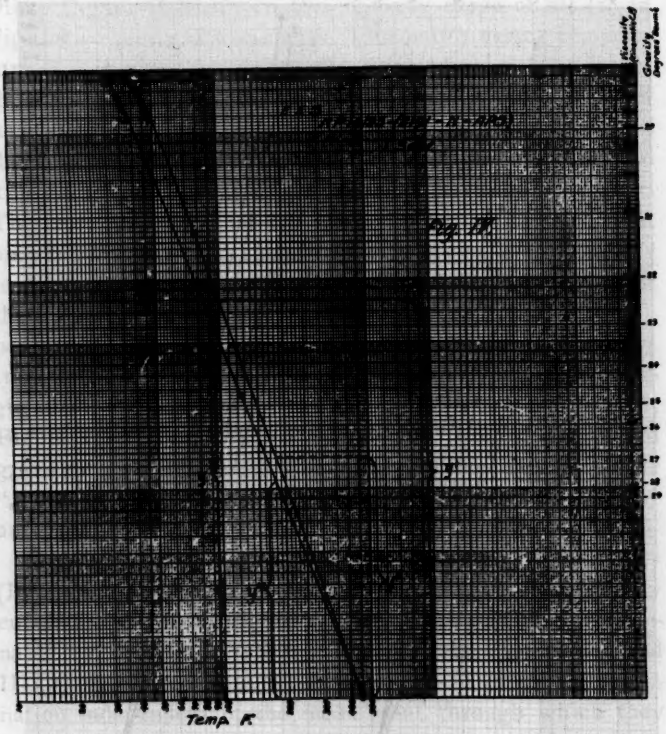
II.

Referring to Figure IV, there are shown the viscosity and gravity curves of a fuel oil, plotted to the scales described


Temp. Scale Range for III.
Temp. Scale Range for III a.

above. It is obvious that, given *one* viscosity and *one* gravity determination, and using the mean point A, the curves can be plotted, and their relation analyzed by inspection.

The mathematical relation of the curves of Figure IV is



simple. y and y' are the ordinates, of the gravity and viscosity curves respectively; for the same temperature, 100 degrees F., the values of x and x' are identical; though, of course, if the gravity and viscosity are obtained at different

temperatures, the actual values of x and x' must be used to obtain the respective slopes of the curves according to the typical equation.

$$\frac{\log y}{\log x} = n.$$

The theory of the curves shown in Figure IV and the use of the tables can be demonstrated by assigning numerical values to the curves of Figure IV and finding one unknown quantity, say viscosity at 180 degrees F.

(*a*) n, the slope of the gravity curve, is equal to $\frac{\log y}{\log x}$

(*b*) n', the slope of the viscosity curve, is equal to $\frac{\log y'}{\log x'}$

(1) In Figure IV, y is the *gravity* ordinate; y' the *viscosity* ordinate at 100 degrees F.

(2) Let y''' and y'' represent, respectively, the gravity and viscosity ordinates at 180 degrees F.

Assigning numerical values, let

$$\left.\begin{array}{l} y = 23.3 \text{ degrees Bé:} \\ y' = 80 \text{ (Vis. } c.\ p.\text{)} \end{array}\right\} x = x' = 100 \text{ degrees F.}$$

The first step is to find the ratio of the slopes, that is, $\frac{n'}{n}$

This can be found from equations (*a*) and (*b*), as follows, remembering that in this selected case $x = x'$:

$$(3)\quad \frac{\log y'}{\log y} = \frac{n'}{n} \times \frac{\log x'}{\log x} = \frac{n'}{n}$$

To the logarithmic scale used, $\left\{\begin{array}{l} \log y' \text{ (viscosity } c.p.\text{)} = 1.903 \\ \log y \text{ (gravity Bé)} = 1.719 \end{array}\right.$

$$\therefore \frac{n'}{n} = 1.107 = \text{ratio of the slopes of the curve (4).}$$

Now, referring back to (2): If the gravity at 180 degrees F. is 27.8 degrees Bé, what is the viscosity at that temperature?

$y''' = 27.8$ degrees Bé; from the table (gravity column)

$\log y''' = 1.043$.

$$\frac{\log y''}{\log y'''} = \frac{n'}{n} \quad \frac{\log x}{\log x'} = \frac{n'}{n} \quad (x = x')$$

$$\log y'' = \frac{n'}{n} \log y''' = 1.107 \times 1.043$$

$$\log y'' = 1.155 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Consulting the table, under the *viscosity* column, log y'' corresponds to 14.3 (kinematic), or, under the Saybolt Universal column, to 78 seconds. These values check with the experimentally determined values, at 180 degrees F., of the oil whose curves are shown in Figure IV.

III.

All of the above equations may be used to construct a table covering a wide range of values, from which given x, x', y, y' and y''', their logs may be taken out by inspection, interpolating as necessary, for the solution of the equation

$$\log y'' = \frac{n'}{n} \log'''.$$

Such a table is illustrated in Figure V.

Column 1 gives the Baumé Gravity scale. Opposite each value, in Column 2 is shown the value of "Log y."

Columns 3, 4, 5 and 6 give the Kinematic, Saybolt Furol (seconds), Saybolt Universal (seconds) and Engler Viscosity Scales; and, further, parallel readings are equivalent to each other. Column 7 gives the value of log y for viscosity values and may be used for any functions of viscosity in the several equations. Interpolations are necessary as in any table; further values above and below the range given may be found by selecting logs, or anti-logs, for multiples or factors, paying attention to the characteristic in arithmetical or slide rule use of the logs.

Columns 8 and 9 give respectively the values of x and x', which are temperatures F. and their corresponding logarithms. (In this connection, it is to be observed that *all* values shown in this table take into account the logarithmic

VISCOSITY - GRAVITY - TEMPERATURE

VISCOSITY - TEMPERATURE

GRAVITY - TEMPERATURE

GRAVITY		VISCOSITY					VISCOSITY						
Degrees Baumé	Log γ	Ubbe-lohde o.-b.	Saybolt Furol (Seconds)	Saybolt Universal (Seconds)	Degrees Engler	log γ' (5)	Ubbe-lohde o.-b.	Saybolt Furol (Seconds)	Saybolt Universal (Seconds)	Degrees Engler	log γ' (5)	Temp. F. degrees (2)	log T (2)
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
20.0	0.000	2.0		32.0	1.00	0.477	70	30	200	2.4	1.041	60	0.079
20.0	0.017	2.0		32.0	1.00	0.544	72	31	205	2.5	1.057	62	0.093
20.0	0.033	2.0		32.0	1.00	0.602	74	32	210	2.6	1.072	64	0.107
20.0	0.050	2.0		32.0	1.00	0.653	76	33	215	2.7	1.087	66	0.121
20.0	0.066	2.0		32.0	1.00	0.699	78	34	220	2.8	1.102	68	0.134
20.0	0.083	2.0		32.0	1.00	0.740	80	35	225	2.9	1.116	70	0.147
20.0	0.099	2.0		32.0	1.00	0.778	82	36	230	3.0	1.130	72	0.160
20.0	0.116	2.0		32.0	1.00	0.813	84	37	235	3.1	1.144	74	0.172
20.0	0.132	2.0		32.0	1.00	0.845	86	38	240	3.2	1.158	76	0.184
20.0	0.149	2.0		32.0	1.00	0.875	88	39	245	3.3	1.172	78	0.196
20.0	0.165	2.0		32.0	1.00	0.903	90	40	250	3.4	1.185	80	0.208
20.0	0.182	2.0		32.0	1.00	0.929	92	41	255	3.5	1.198	82	0.219
20.0	0.198	2.0		32.0	1.00	0.954	94	42	260	3.6	1.211	84	0.230
20.0	0.215	2.0		32.0	1.00	0.978	96	43	265	3.7	1.224	86	0.241
20.0	0.231	2.0		32.0	1.00	1.000	98	44	270	3.8	1.237	88	0.252
20.0	0.248	2.0		32.0	1.00	1.021	100	45	275	3.9	1.250	90	0.262
20.0	0.264	2.0		32.0	1.00	1.041	102	46	280	4.0	1.263	92	0.272
20.0	0.281	2.0		32.0	1.00	1.060	104	47	285	4.1	1.275	94	0.282
20.0	0.297	2.0		32.0	1.00	1.078	106	48	290	4.2	1.288	96	0.292
20.0	0.314	2.0		32.0	1.00	1.095	108	49	295	4.3	1.300	98	0.301
20.0	0.330	2.0		32.0	1.00	1.112	110	50	300	4.4	1.312	100	0.310
20.0	0.347	2.0		32.0	1.00	1.128	112	51	305	4.5	1.324	102	0.319
20.0	0.363	2.0		32.0	1.00	1.144	114	52	310	4.6	1.336	104	0.328
20.0	0.380	2.0		32.0	1.00	1.159	116	53	315	4.7	1.348	106	0.337
20.0	0.396	2.0		32.0	1.00	1.174	118	54	320	4.8	1.360	108	0.345
20.0	0.413	2.0		32.0	1.00	1.188	120	55	325	4.9	1.372	110	0.353
20.0	0.429	2.0		32.0	1.00	1.202	122	56	330	5.0	1.384	112	0.361
20.0	0.446	2.0		32.0	1.00	1.216	124	57	335	5.1	1.396	114	0.369
20.0	0.462	2.0		32.0	1.00	1.230	126	58	340	5.2	1.408	116	0.377
20.0	0.479	2.0		32.0	1.00	1.243	128	59	345	5.3	1.420	118	0.385
20.0	0.495	2.0		32.0	1.00	1.256	130	60	350	5.4	1.432	120	0.392
20.0	0.512	2.0		32.0	1.00	1.269	132	61	355	5.5	1.444	122	0.400
20.0	0.528	2.0		32.0	1.00	1.282	134	62	360	5.6	1.456	124	0.407
20.0	0.545	2.0		32.0	1.00	1.295	136	63	365	5.7	1.468	126	0.414
20.0	0.561	2.0		32.0	1.00	1.307	138	64	370	5.8	1.480	128	0.421
20.0	0.578	2.0		32.0	1.00	1.320	140	65	375	5.9	1.492	130	0.428
20.0	0.594	2.0		32.0	1.00	1.332	142	66	380	6.0	1.504	132	0.435
20.0	0.611	2.0		32.0	1.00	1.344	144	67	385	6.1	1.516	134	0.442
20.0	0.627	2.0		32.0	1.00	1.356	146	68	390	6.2	1.528	136	0.449
20.0	0.644	2.0		32.0	1.00	1.368	148	69	395	6.3	1.540	138	0.456
20.0	0.660	2.0		32.0	1.00	1.380	150	70	400	6.4	1.552	140	0.463
20.0	0.677	2.0		32.0	1.00	1.392	152	71	405	6.5	1.564	142	0.470
20.0	0.693	2.0		32.0	1.00	1.404	154	72	410	6.6	1.576	144	0.477
20.0	0.710	2.0		32.0	1.00	1.416	156	73	415	6.7	1.588	146	0.484
20.0	0.726	2.0		32.0	1.00	1.428	158	74	420	6.8	1.600	148	0.491
20.0	0.743	2.0		32.0	1.00	1.440	160	75	425	6.9	1.612	150	0.498
20.0	0.759	2.0		32.0	1.00	1.452	162	76	430	7.0	1.624	152	0.505
20.0	0.776	2.0		32.0	1.00	1.464	164	77	435	7.1	1.636	154	0.512
20.0	0.792	2.0		32.0	1.00	1.476	166	78	440	7.2	1.648	156	0.519
20.0	0.809	2.0		32.0	1.00	1.488	168	79	445	7.3	1.660	158	0.526
20.0	0.825	2.0		32.0	1.00	1.500	170	80	450	7.4	1.672	160	0.533
20.0	0.842	2.0		32.0	1.00	1.512	172	81	455	7.5	1.684	162	0.540
20.0	0.858	2.0		32.0	1.00	1.524	174	82	460	7.6	1.696	164	0.547
20.0	0.875	2.0		32.0	1.00	1.536	176	83	465	7.7	1.708	166	0.554
20.0	0.891	2.0		32.0	1.00	1.548	178	84	470	7.8	1.720	168	0.561
20.0	0.908	2.0		32.0	1.00	1.560	180	85	475	7.9	1.732	170	0.568
20.0	0.924	2.0		32.0	1.00	1.572	182	86	480	8.0	1.744	172	0.575
20.0	0.941	2.0		32.0	1.00	1.584	184	87	485	8.1	1.756	174	0.582
20.0	0.957	2.0		32.0	1.00	1.596	186	88	490	8.2	1.768	176	0.589
20.0	0.974	2.0		32.0	1.00	1.608	188	89	495	8.3	1.780	178	0.596
20.0	0.990	2.0		32.0	1.00	1.620	190	90	500	8.4	1.792	180	0.603
20.0	1.007	2.0		32.0	1.00	1.632	192	91	505	8.5	1.804	182	0.610
20.0	1.023	2.0		32.0	1.00	1.644	194	92	510	8.6	1.816	184	0.617
20.0	1.040	2.0		32.0	1.00	1.656	196	93	515	8.7	1.828	186	0.624
20.0	1.056	2.0		32.0	1.00	1.668	198	94	520	8.8	1.840	188	0.631
20.0	1.073	2.0		32.0	1.00	1.680	200	95	525	8.9	1.852	190	0.638
20.0	1.089	2.0		32.0	1.00	1.692	202	96	530	9.0	1.864	192	0.645
20.0	1.106	2.0		32.0	1.00	1.704	204	97	535	9.1	1.876	194	0.652
20.0	1.122	2.0		32.0	1.00	1.716	206	98	540	9.2	1.888	196	0.659
20.0	1.139	2.0		32.0	1.00	1.728	208	99	545	9.3	1.900	198	0.666
20.0	1.155	2.0		32.0	1.00	1.740	210	100	550	9.4	1.912	200	0.673
20.0	1.172	2.0		32.0	1.00	1.752	212	101	555	9.5	1.924	202	0.680
20.0	1.188	2.0		32.0	1.00	1.764	214	102	560	9.6	1.936	204	0.687
							216	103	565	9.7	1.948	206	0.694
							218	104	570	9.8	1.960	208	0.701
							220	105	575	9.9	1.972	210	0.708
							222	106	580	10.0	1.984	212	0.715
							224	107	585	10.1	1.996	214	0.722
							226	108	590	10.2	2.008	216	0.729
							228	109	595	10.3	2.020	218	0.736
							230	110	600	10.4	2.032	220	0.743
							232	111	605	10.5	2.044	222	0.750
							234	112	610	10.6	2.056	224	0.757
							236	113	615	10.7	2.068	226	0.764
							238	114	620	10.8	2.080	228	0.771
							240	115	625	10.9	2.092	230	0.778
							242	116	630	11.0	2.104	232	0.785
							244	117	635	11.1	2.116	234	0.792
							246	118	640	11.2	2.128	236	0.799
							248	119	645	11.3	2.140	238	0.806
							250	120	650	11.4	2.152	240	0.813
							252	121	655	11.5	2.164	242	0.820
							254	122	660	11.6	2.176	244	0.827
							256	123	665	11.7	2.188	246	0.834
							258	124	670	11.8	2.200	248	0.841
							260	125	675	11.9	2.212	250	0.848
							262	126	680	12.0	2.224	252	0.855
							264	127	685	12.1	2.236	254	0.862
							266	128	690	12.2	2.248	256	0.869
							268	129	695	12.3	2.260	258	0.876
							270	130	700	12.4	2.272	260	0.883
							272	131	705	12.5	2.284	262	0.890
							274	132	710	12.6	2.296	264	0.897
							276	133	715	12.7	2.308	266	0.904
							278	134	720	12.8	2.320	268	0.911
							280	135	725	12.9	2.332	270	0.918
							282	136	730	13.0	2.344	272	0.925
							284	137	735	13.1	2.356	274	0.932
							286	138	740	13.2	2.368	276	0.939
							288	139	745	13.3	2.380	278	0.946
							290	140	750	13.4	2.392	280	0.953
							292	141	755	13.5	2.404	282	0.960
							294	142	760	13.6	2.416	284	0.967
							296	143	765	13.7	2.428	286	0.974
							298	144	770	13.8	2.440	288	0.981
							300	145	775	13.9	2.452	290	0.988
							302	146	780	14.0	2.464	292	0.995
							304	147	785	14.1	2.476	294	1.002
							306	148	790	14.2	2.488	296	1.009
							308	149	795	14.3	2.500	298	1.016
							310	150	800	14.4	2.512	300	1.023
							312	151	805	14.5	2.524	302	1.030
							314	152	810	14.6	2.536	304	1.037
							316	153	815	14.7	2.548	306	1.044
							318	154	820	14.8	2.560	308	1.051
							320	155	825	14.9	2.572	310	1.058
							322	156	830	15.0	2.584	312	1.065
							324	157	835	15.1	2.596	314	1.072
							326	158	840	15.2	2.608	316	1.079
							328	159	845	15.3	2.620	318	1.086
							330	160	850	15.4	2.632	320	1.093
							332	161	855	15.5	2.644	322	1.100
							334	162	860	15.6	2.656	324	1.107
							336	163	865	15.7	2.668	326	1.114
							338	164	870	15.8	2.680	328	1.121
							340	165	875	15.9	2.692	330	1.128
							342	166	880	16.0	2.704	332	1.135
							344	167	885	16.1	2.716	334	1.142
							346	168	890	16.2	2.728	336	1.149
							348	169	895	16.3	2.740	338	1.156
							350	170	900	16.4	2.752	340	1.163
							352	171	905	16.5	2.764	342	1.170
							354	172	910	16.6	2.776	344	1.177
							356	173	915	16.7	2.788	346	1.184
							358	174	920	16.8	2.800	348	1.191
							360	175	925	16.9	2.812	350	1.198
							362	176	930	17.0	2.824	352	1.205
							364	177	935	17.1	2.836	354	1.212
							366	178	940	17.2	2.848	356	1.219
							368	179	945	17.3	2.860	358	1.226
							370	180	950	17.4	2.872	360	1.233
							372	181	955	17.5	2.884	362	1.240
							374	182	960	17.6	2.896	364	1.247
							376	183	965	17.7	2.908	366	1.254
							378	184	970	17.8	2.920	368	1.261
							380	185	975	17.9	2.932	370	1.268
							382	186	980	18.0	2.944	372	1.275
							384	187	985	18.1	2.956	374	1.282
							386	188	990	18.2	2.968	376	1.289
							388	189	995	18.3	2.980	378	1.296
							390	190	1000	18.4	2.992	380	1.303
							392	191	1005	18.5	3.004	382	1.310
							394	192	1010	18.6	3.016	384	1.317
							396	193	1015	18.7	3.028	386	1.324
							398	194	1020	18.8	3.040	388	1.331
							400	195	1025	18.9	3.052	390	1.338
							402	196	1030	19.0	3.064	392	1.345
							404	197	1035	19.1	3.076	394	1.352
							406	198	1040	19.2	3.088	396	1.359
							408	199	1045	19.3	3.100	398	1.366
							410	200	1050	19.4	3.112	400	1.373

(1)

VISCOSITY - GRAVITY - TEMPERATURE

$$\frac{\log \gamma}{\log \gamma'} = \phi \quad (1)$$

$$\frac{\log \gamma'}{\log \gamma''} = \phi' \quad (2)$$

(Where Vis. and Gravity determinations are at the same temperature, $\phi = \phi'$

$$\log \gamma'' = \frac{\phi'}{\phi} \cdot \log \gamma'' \quad (3)$$

Where γ'' is the gravity for which it is required to find the corresponding viscosity.

(3)

VISCOSITY OR GRAVITY

Given Viscosity or Gravity γ at temperature t, find Viscosity or Gravity γ' at temperature t'.

$$\log \gamma' = \frac{\log \gamma \log \gamma''}{\log \gamma} \quad (4)$$

$$\gamma'' = \gamma_0 (o.-b.) + 0.00 \gamma_0 - \frac{100}{\gamma_0}$$

| | | |

scale upon which it is based, and in particular the abscissae are all referred to the "mean point". Therefore, the common logarithms of the numerical values of x (x') shown in Column 8, differ from the log values of Column 9, and cannot be used.)

It now remains to demonstrate the use of the table.

1. Given a sample of fuel oil whose viscosity and gravity at 122 degrees F. are, respectively, 78 seconds Saybolt Furol and 20.0 degrees Baumé.

If the gravity at 210 degrees F. is 24.15, what is the viscosity at that temperature?

(Col. 1) $y = 20$ degrees Bé

From Col. 2, $\log y = 2.699$

(Col. 8) $x = 122$ degrees F.

From Col. 9, $\log x = 0.569$ (By interpolation)

$$n = \frac{\log y}{\log x} = 4.74 = \text{Slope of curve}$$

(Col. 4) $y' = 78$ seconds Furol

From Col. 7, $\log y' = 2.214$

$x' = 122$ degrees F.

$\log x' = 0.569$

$$n' = \frac{\log y'}{\log x'} = \frac{2.214}{0.569} = 3.89 = \text{Slope of curve.}$$

From equation (1)

$$\log y'' = \frac{n'}{n} \log y'''$$

$y''' = 24$ degrees .15 Bé

From Col. 2, $\log y''' = 1.553$ (By interpolation)

$$\therefore \log y'' = \frac{389}{474} \times 1.553 = 1.485.$$

$y'' = 30.5$ V_k (By interpolation)

(30.5 V_k is equivalent to 19 seconds Saybolt Furol.)

IV.

Additional Valuable Uses of the Table.

The Approximate Viscosity and for Gravity of a Fuel Oil at any temperature (below the limit of 220 degrees F.) can readily be determined, given its viscosity and / or gravity at another temperature; likewise temperatures can be determined, given the second viscosity and / or gravity. Mathematically, it is a case of simple proportion.

Using the same coordinates as above, ordinates represent viscosity and gravity values, abscissae represent temperatures. The values are taken out, in the table, as described above. The illustrations below are those of oils tested in laboratory apparatus. Since the tests were routine, the experimental results are probably subject to an error of at least ± 5 per cent.

EXAMPLE I.

Given Viscosity 111 seconds Saybolt Furol at 77 degrees F.
Find Viscosity at 180 degrees F.
By proportion

$$\log y' = \frac{\log y \log x}{\log x'}$$

$y = 111$; $\log y = 2.364$
$x = 77$ degrees; $\log x = 0.769$
$x' = 180$ degrees; $\log x' = 0.397$

$\therefore \log y' = 1.220$
$y' = 16.6$ $V_k = 14$ seconds Saybolt Furol.

Note: This checks with the experimental determination.

EXAMPLE II.

Given Viscosity 121 seconds Saybolt Universal at 77 degrees F.
Find Viscosity at 122 degrees F.
$\log y' = 1.036$ $y = 121$; $\log y = 1.38201$
$y' = 10.86$ $V_k = 64$ seconds Saybolt Universal.

$x = 77$ degrees; $\log x = 0.769$

$x' = 122$ degrees; $\log x' = 0.569$

Note: Experimental determination was 61 seconds.

Error = 4.7 per cent.

A large number of determinations made as shown above, and compared with laboratory results, using the samples, indicates that the table may be used to predict viscosities with an indicated accuracy of ± 5 per cent. As stated before, since the laboratory determinations are of a routine nature, they are doubtless subject to some error.

Specific Gravity may be treated as just shown, in which case, $\log y$ and $\log y'$ are taken out from Column 2.

The equation

$$\log y' = \frac{\log y \log x}{\log x'}$$

can, of course, be solved for any factor, the other three being given, using the Table as described. For example, if the viscosity of an oil at one temperature is known, and it is desired to find the temperature necessary to reduce the oil to a given viscosity (Engler, Saybolt Furol or Saybolt Universal), the equation becomes

$$\log x' = \frac{\log y \log x}{\log y'},$$

y' in this case being the viscosity to which it is desired to reduce the oil.

EXAMPLE.

A fuel oil has a viscosity of 80 seconds Saybolt (Furol) at 77 degrees F. What is the temperature necessary to reduce the oil at 3 degrees Engler?

$$\log x' = \frac{1.301 \times 0.769}{2.225} = 0.449$$

$x = 160$ degrees F.

$y' = 3$ degrees E. $\log y = 1.301$

$x = 77$ $\log x = 0.769$

$y = 80$ seconds (Furol) $\log y = 2.225$

THE TREND OF AIRCRAFT ENGINE DEVELOPMENT.

By COMMANDER E. E. WILSON, U. S. N., MEMBER.

From the very beginning of aircraft engine development the ratio of the engine weight to its power output has been the measure of excellence. Efforts have been directed toward reducing this ratio because the weight of the power plant itself is reflected back into the gross weight of the airplane. The heavier the engine for a given power, the greater the wing area necessary to support it. This, of course, results in greater structure weight which in turn requires more power for a given performance. These factors have forced our engines from a dry weight of about 6 pounds per horsepower and an output of about 30 H.P. to a dry weight of 1.4 per H.P. and an output of 800 H.P.

As long as we are comparing watercooled engines with each other and aircooled engines with each other, the dry weight of the engine is a reasonable basis for comparison, but when we compare aircooled and watercooled engines, we find this dry weight basis no longer useful. The weight of the cooling system is roughly .7 of a pound per horsepower and aircooled engines have the advantage of reduction in power plant weight by this amount. Our comparisons are therefore based on the total power plant weight rather than the old dry weight.

In order to meet the competition of the lighter aircooled engines, watercooled engines have been speeded up. Thus, the Packard 2A-1500, which has a normal power output of 500 H.P. at 2100 R.P.M., has recently completed a fifty-hour full throttle test developing 600 H.P. at 2500 R.P.M. This increase in power is sufficient to overcome the additional weight

of gearing and cooling system with the result that high speed watercooled engines are now on a parity with the lower speed aircooled engines insofar as the weight per horsepower of the total power plant is concerned.

The aircooled engines, however, still have an advantage. A careful analysis of engine faults shows that about 40 per cent of the total mechanical failures are due to troubles with the fuel, oil and water lines; that is, the so-called "plumbing" system. Of this 40 per cent of mechanical faults, about one-third is due to the cooling system. By eliminating the cooling system through the use of aircooled engines, we can immediately eliminate a large percentage of the mechanical failures.

The aircooled engines' reply to the challenge of the watercooled engines through high crankshaft speeds is an increase in its own crankshaft speed. Present aircooled engines are for the most part of the radial type and the problem of turning radials at higher speeds is more complex than is that of watercooled engines. It seems likely, however, that radial engines will follow suit and the race is again on insofar as the weight/horsepower ratio is concerned.

Closely tied up with the weight of power plant is fuel economy. A modern aircraft engine will, in three hours of full throttle flying, consume practically its own weight in fuel. Obviously, improved economy offers the greatest promise in weight reduction at the present time. Watercooled engines are at present superior in economy to aircooled engines because of better cooling and better design. New aircooled engines incorporating rotary induction by means of a gear-driven fan and having greatly improved cylinder cooling give promise of narrowing the gap between the economy of air and watercooled engines.

Other things being equal, improved economy can be had in all engines through the use of high compression ratios. At present the compression ratio is limited by the character of the fuel. Ordinary aviation gasoline cannot be run without

detonation at compression ratios of over 5½ to 1 in the present engines. Gasolines from different fields have greater or less tendencies to detonation. The tendency to detonation is a function of the physical properties of the gasolines. Gasolines containing a high percentage of aromatics permit the use of higher compression ratios. From this viewpoint, gasolines from some of the California fields are superior to gasolines from some of the eastern fields.

There are a number of different substances which can be blended with aviation gasoline to reduce the tendency to detonation. Alcohol is one of these and benzol is another. We have successfully used a blend of alcohol, benzol and gasoline for this purpose and have attained higher compression ratios through this blend. However, our experience has been that the tendency of alcohol to absorb atmospheric moisture, which in turn results in a separation of the constituents of the blend, makes this fuel not entirely satisfactory. Benzol at low temperatures will crystallize out and clog the fuel system. It is therefore not entirely suited to cold weather operation or operation at high altitudes.

Ethyl Fluid which has as its base tetraethyl lead is an excellent anti-knock compound. It can be blended with aviation gasoline and when so blended will permit the use of much higher compression ratios. Since, however, it has a lead base, precautions must be taken in its use to avoid lead poisoning. Experience to date indicates that there is no hazard in its use in aircraft where the propeller slipstream results in very great dilution and that far more danger is to be encountered from carbon monoxide in the exhaust gases. Ethyl Fluid has been used in naval aircraft with considerable success but the real requirement is for a fuel which need not be blended or doped.

Considerable work has been done recently in the field of cracked gasolines. These fuels not only have the advantage that a much higher yield of gasoline can be had from a given crude but they will definitely permit an increase in com-

pression ratio to about 6:1. The only difficulty that has been encountered in the use of this fuel lies in the fact that all fuels submitted to date deposit a gummy substance on the intake valves which interferes with their operation. Efforts are being made to eliminate this defect and there is every hope that a solution will be forthcoming. Once a suitable fuel is had, improved economy will result and thus reduce the weights chargeable to the power plant.

Tied up very definitely with economy is propeller efficiency. It has been stated that the rough measure of propeller speed versus speed of advance of the airplane is had in the factor 10 revolutions per minute per mile per hour. In other words, for a speed advance of 100 miles per hour, the propeller for best efficiency should turn about 1000 R.P.M. Modern aircraft engines when direct-driven are normally operated at about 1800 R.P.M. at full throttle for slow speed aircraft. In high speed aircraft of the order of 200 miles per hour, direct-drive engines turning at 2000 R.P.M. can be economically used. For the slow heavy duty aircraft, reduction gears must be used for maximum economy. Reduction gears also have the great advantage of increased thrust at low speed. These are the factors which were incorporated in the two *PN-9's* for the West Coast-Hawaiian Flight in which the engines turned about 2300 R.P.M. in level flight at full speed with the propeller turning at about 1150 R.P.M. For a speed of 120 miles per hour, this was an excellent combination. The success of this combination is apparent when we realize that the two *PN-9's* have a higher percentage of useful to full load than any other airplanes of which we know. With the useful load equal approximately to their dead load, they were, through the high thrust available, able to get off quickly and to cruise very economically throughout their whole range. These airplanes employ the Packard 1A-1500 engines with 2 to 1 reduction gears previously mentioned. An increase in propeller efficiency of about 9 per cent resulted from the lower propeller speed.

The matter of economy is of greatest importance in long range aircraft. The specific fuel consumption varies with the throttle opening, the form of the curve being influenced largely by carburetor design and the friction horsepower of the engine, Figure 1. Thus, for a specific fuel consumption of .51 at full

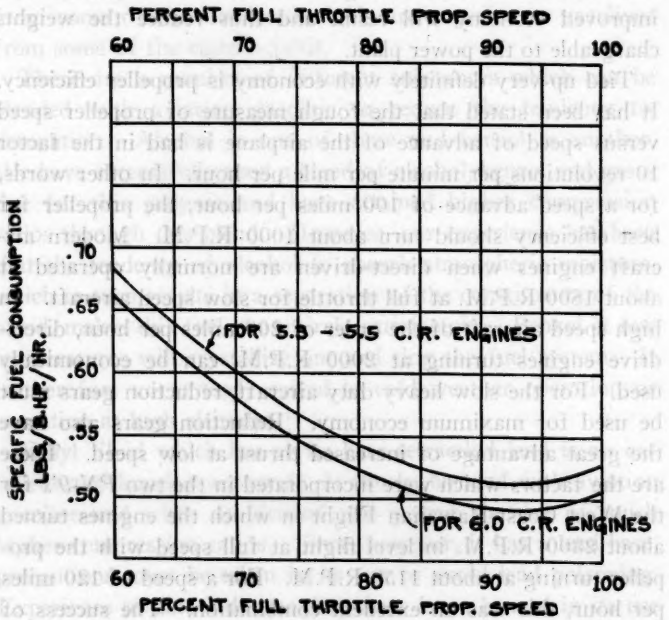


FIG. 1

CHARACTERISTIC SPECIFIC FUEL CONSUMPTION CURVE OF AIRCRAFT ENGINES ON PROPELLER LOAD.

throttle, the fuel consumption at part throttle on a propeller load goes to a minimum of about .48 at about 93 per cent of full throttle propeller speed. It then rises again crossing the .51 position at about 85 per cent full throttle. Below this position, it increases quite rapidly.

The power required to drive the airplane is a function of the angle of attack. The heavier the load of the aircraft the greater the angle of attack for a given speed. There is an angle at which the lift-drag ratio of a given wing is a maximum. By combining the curves of engine economy and airplane performance, it is possible to calculate the economical speed for a given loading. In the case of the *PB-1*, another airplane designed for the West Coast-Hawaiian Flight, using two Packard 800 H.P. engines with a gear ratio of 2:1 and weighing fully loaded 28,000 pounds, the economical speed after take-off is about 87½ knots. The economical speed at the end of the flight when the airplane weighs about 14,000 pounds is 66 knots. In other words, a change in economical speed of 21½ knots is had from the beginning to the end of a long flight. By calculating the fuel consumption at these different speeds, it is possible to make out a table showing that the air speed should be reduced roughly one knot every 2½ hours. For maximum range, this table must be followed carefully.

The above calculations are based upon flying in still air. The conditions are changed in case an adverse or favorable wind is encountered. Against an adverse wind, it is necessary to fly at high speeds in order to reduce the time during which the wind will have its effect. For favorable winds, it is necessary to reduce the air speed to increase the time during which the wind is effective. This increase and reduction can be calculated taking into consideration the engine fuel consumption curve and the attitude of the airplane at the given load and a curve plotted which will give the best air speed for maximum range.

In our calculations, we must differentiate between *range* and *duration*. Maximum *duration* can be had by flying just sufficiently above the stalling speed to maintain control, but this maximum duration will not result in maximum range unless exactly the right favorable wind is had. Maximum *range* is

had by flying at approximately 1½ times the stalling speed. In any long range flight, it is necessary to appreciate these conditions fully and to utilize them to the very best advantage.

Aircraft engines require a different mixture ratio of air to gasoline for maximum power and maximum economy. Since aircraft require a maximum power for take-off, the carburetors are ordinarily adjusted to give a rich mixture at this time. Once the engine is throttled, this mixture will be too rich for best economy and it becomes necessary to lean it out. This is accomplished by a manually operated Mixture Control through which the conditions can be adjusted for best economy. In case the mixture is leaned too far, "after-burning" will result and due to slight inequalities in distribution the engine will begin to "pop." The mixture can then be richened until the popping just ceases at which point best economy will be had. Leaning out the mixture results in the falling off in crank speed of the order of about 50 R.P.M. A pilot on a long range flight must therefore so adjust his engine speed that the airplane is flying at the prescribed air speed with the mixture control at the position for maximum economy. In fact, it is customary to throttle to the engine speed which will give about thirty revolutions more than are required for a given air speed. The mixture is then leaned through the use of the mixture control until about thirty revolutions are lost, whereupon the airplane is settled down to its cruising speed at best economy. Obviously, on a flight involving 25 or 30 hours, and with continual changes in speed and engine adjustment a pilot must be alert to get the maximum out of his plane and must have information as to the effective wind velocity in order to establish the proper air speed. All these factors were known and employed by the pilots who essayed the West Coast-Hawaiian Flight.

It is with a full appreciation of the factors involved in this weight per horsepower ratio that aircraft engine development in the Navy has been fostered. These considerations pertain

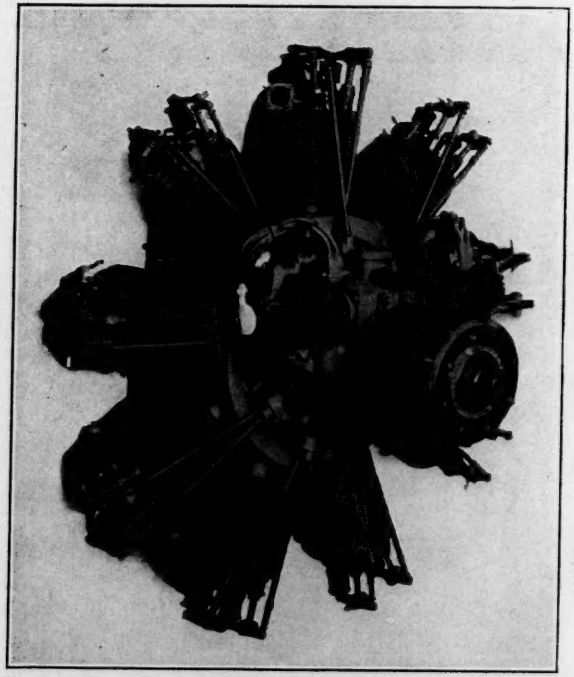


FIG. 2.—WRIGHT MODEL J-4A, 200 H. P. AIRCOOLED ENGINE.

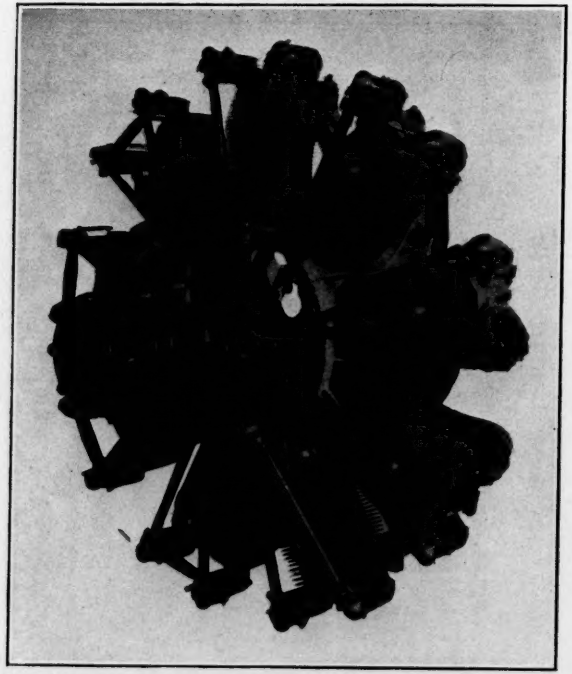


FIG. 3.—WRIGHT MODEL P-2, 450 H. P. AIRCOOLED ENGINE.

FIG. 4.—WRIGHT MODEL R-1200, 350 H. P. AIRCOOLED ENGINE.

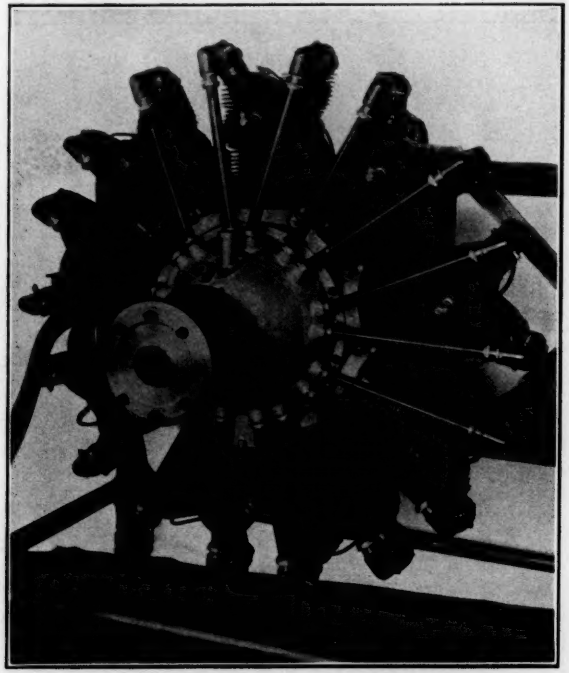


FIG. 5.—PRATT AND WHITNEY MODEL R-1300, 375 H. P. AIRCOOLED ENGINE.

primarily to the power plant but there are certain other factors which, in conjunction with the operation problem, have influenced power plant development.

Watercooled engines to date have been chiefly of the in-line type, in single rows of six cylinders, in double rows with a total of eight of twelve cylinders, and in three or four rows with a total of twelve cylinders. Our aircooled engines have, for the most part, been radials with nine cylinders located around the single crankshaft. It is perfectly feasible to operate an in-line engine as an aircooled engine. One of the recent examples of this is the aircooled Liberty developed by the Air Service Engineering Division at McCook Field. Any in-line aircooled engine must, however, of necessity be longer or higher than a watercooled engine of similar displacement. The Liberty lends itself well to aircooling for the reason that the distance between cylinder centers in this engine is much larger than in later watercooled engines. Later and more compact engines do not so lend themselves. This factor of length is of the very greatest importance in Naval Aircraft and is one of the factors which has dictated the Navy's development of the radial engine.

A fundamental requirement of naval aircraft is that they be as light as possible and as small as possible for the given performance. Most men-of-war are limited in the space which is available for aircraft on deck and the problem of handling heavy weights in a sea way or of catapulting heavy weights is a difficult one. This has forced us into the endeavor to attain in very small airplanes with small engines as much performance as can be had in landplanes of larger size with larger engines. The Vought two-seaters in use on battleships and light cruisers are aircraft of this type. The Navy is then forced to keep down the size and weight of the structure to a much greater degree than is necessary on shore.

The same restriction applies to a lesser degree on carriers. The restriction does not so much concern the smaller types of

airplanes but in the heavier bombers the wing span is definitely limited so that an airplane designed around the large water-cooled engine is not a good job aerodynamically within the wing span limits of even the large carriers.

Another fundamental requirement for carrier use and in fact for all aircraft is that of vision. The vision ahead in water-cooled engines is ordinarily poor. It is possible, however, in the much shorter radial engines, to locate the pilot close up behind the engine, thus giving him excellent vision all around not only for deck landing but for combat. The small overall length of the radial makes it a very desirable engine for naval use and it is on this basis that we have gone ahead with our radial development even though the in-line aircooled will have certain advantages.

The Navy's development of the radial has proceeded in the face of continued opposition. Watercooled engine builders point to the fact that the watercooled "V" engine has small frontal area and offers less resistance. They seldom take into consideration the fact that the fuselage must be of a reasonable size and that the masking effect of this fuselage accounts for the great portion of the frontal area of the aircooled engine. They also neglect the drag area of the radiators required for watercooled engines. As a matter of fact the radial can be so cowled in that only the cylinder heads offer a drag area in excess of the masking effect of the fuselage and this area is not greater than that offered by radiators.

On the basis of our weight limitations, the Navy has fostered the development of a line of three aircooled engines. The first of these is the Wright Model "J" (Figure 2), a nine-cylinder static radial of 200 H.P. and weighing dry about 500 pounds. This engine has a piston displacement of about 800 cubic inches. The second development was the Wright Model "P-2," a nine-cylinder static radial developing 455 H.P. and weighing dry about 825 pounds. This engine has a capacity of 1650 cubic inches, practically the same as the Liberty. The

Wright Model "J" has been employed in two-seat observation and single-seat fighters for battleships and light-cruisers. It has also been employed in training planes. To date, this engine has proved most satisfactory. Ships returning from the Australian Cruise have reported a life between overhauls of considerably over 200 hours as compared with the average life between overhauls of about 75 hours for the Liberty. Two engines of this type have been reported with over 275 hours without loss in power and without major overhaul. So successful has this engine been in the Navy that it is now coming into wide use commercially. It has been used in a number of Huff-Daland airplanes for dusting cotton crops and fruit trees in the south. It has been used in South America, Canada, and Cuba with great success. In the three-engined Fokker commercial transport, these engines have been most successful and it seems likely that future commercial work in this country will be based around this engine which was developed entirely by the Navy.

The Wright Model "P" (Figure 3), the large radial has passed its dynamometer tests and been flown successfully in an airplane. A newer model incorporating rotary induction, enclosed valve gear, improved cylinder construction, and other desirable features, is now undergoing its tests. This engine is intended for heavy duty work and for this purpose probably will, in the future, be geared.

The necessity for increased performance in shipboard airplanes has resulted in the development of the new Wright R-1200 (Figure 4), and the Pratt & Whitney R-1300 (Figure 5). These engines, as the names imply, have 1200 and 1300 cubic inch capacity, respectively, and weigh about 650 pounds. Since they develop 350 H.P., it is apparent that a gain of 150 H.P. over the Wright Model "J" has been had on an increased weight of 150 pounds; that is, the increased power is had at the cost of one pound per horsepower. The R-1200 has recently passed its dynamometer tests with more complete success

than any engine in our experience. It is now being installed in an airplane for flight testing purposes. It is very similar to the Wright Model P-2 and incorporates a number of improvements.

These engines are intended for shipboard observation and fighting planes of greatly improved performance. About the engine can be built airplanes of this type which will not weigh more or be larger than the present 200 H.P. observation and fighting planes. The new airplanes will come within the catapult limitations and the space limitations of our present equipment but will have materially increased performance. In other words, we now have three sizes of aircooled engines, 800 cubic inches, 1200 cubic inches, and 1650 cubic inches capacity about which all of our naval work can be done. The "J" will remain in the training class, the R-1200 and R-1300 in the higher performance class, or twin-engined jobs, and the "P" in the single-engined heavy duty class. Concentration of our work about three sizes of engines having many accessories interchangeable and being in general of the same type will greatly simplify the maintenance and operating problems and should give us superior performance over anything now in sight elsewhere.

Efforts are continually being made to improve the reliability and durability of aircraft engines. The progress mentioned above in the Wright Model "J" is an indication of the work that has been done in increasing durability. It seems not unreasonable to expect life between overhauls of about 300 hours with the Model "J" under reasonable operating conditions. The life between overhauls of the engine depends very largely on the power at which the engine is normally operated. It can be greatly increased by throttling as soon as the flying altitude has been reached. Careful pilots do not drive the engine at a higher output than is required by the mission. This is an angle from the operating viewpoint. Increased durability can be built into the engine and this is handled by purchasing the engines at a much higher rating than that at which they are required to operate. Thus, the Wright Model T-3 engine is

operated with a propeller which will permit a maximum speed of about 1850 R.P.M. at full throttle. The engines are purchased, however, and tested on the block, developing 600 H.P. at 2000 R.P.M. The Packard 1A-1500 will develop in service 500 H.P. at 2100 R.P.M. An engine of this type, as has been stated before, has recently passed a full throttle test developing 600 H.P. at 2500 R.P.M. In other words, we are building into the engine, 50-hour full throttle performance at high speeds and powers in order to have available a much longer operating life at the normal operating speeds and powers.

In the matter of dependability, there are ordinarily two measures of performance. Operating squadrons are accustomed to keep records of the number of forced landings and delayed starts which are encountered in operation. Many of our forced landings were due to faulty ignition. In order to eliminate this, new and improved ignition has been brought forward in the form of Scintilla magnetos with the result that the ignition failures are greatly reduced.

The matter of quick starting is of very great importance, particularly on shipboard aircraft where airplanes must be catapulted on signal, and on carriers where a formation of aircraft may be standing by on deck in an alert condition, ready to take the air on short notice. The failure to start of one airplane in the line will delay all aircraft behind it in the line. It is of paramount importance, therefore, that quick starting be had.

This problem has recently been solved to a very great degree by the Aeromarine inertia starter. In this starter, a flywheel is cranked to a high speed through a gear train. By means of an engaging mechanism, which can be operated at will, the stored energy in the wheel is transmitted to the engine. The great advantage in this starter is that it gives the quick flip to the engine which is required for proper ignition. The inertia starter has been an immediate success and improved types are now in process of development which will weigh considerably less and cost considerably less.

In the matter of dependability and reliability, then, considerable progress has been made. The use of the multi-engine installation, from a viewpoint of reliability, has no great appeal from a military viewpoint. From the commercial angle, multi-engine installations have a particular appeal for passenger carrying. The Fokker three-engined transport is a good example of this type. In the PB-1, two 800 H.P. Packard engines were installed in tandem on the theory that when one engine fails, the airplane can be operated with the other. This is true only to a certain degree, however. When the airplane is fully loaded, it cannot be sustained by one of the engines. Even after half the fuel is gone, the rear engine is not wholly effective because its propeller must be designed to run in the slip-stream of the forward engine and it is therefore inefficient without the forward engine. Where performance is of primary importance, it is more economical to duplicate those accessories which are likely to fail than to duplicate the whole engine installation. To this end, dual ignition is ordinarily fitted.

A very complete series of Trouble Reports are required from the operating air squadrons. These cover not only forced landings and delayed starts, but any form of trouble which may occur. On the basis of these reports, continual refinements and improvements are being made. As previously stated, these reports show that about 40 per cent of the troubles are due to the so-called plumbing system. In the past, the engine has been very carefully engineered and so has the airplane. Less attention has been paid to the installations which have been "mechanized" rather than engineered. These trouble reports have focused attention on this point and great care is being taken now to insure that fuel, oil, and gas lines are made of proper material, that they are properly joined and supported, and that they are so arranged as to avoid damage from vibration. In the matter of dependability, improvement in installations now offers the greatest possibilities. Increased care in this matter will most certainly greatly increase power plant dependability.

To summarize the Navy's aircraft engine development, we find that our ideas are now crystallized around the line of three sizes of aircraft engines of 800, 1200, and 1600 cubic inch capacity which will apparently meet the immediate needs of high performance aircraft for naval aviation. These engines are of the lightest weight per horsepower consistent with dependability and reliability, and in their operation are designed to take advantage of the reduction in power plant weight per horsepower due to improved economy and improved propeller efficiency. Since the organization of the Bureau of Aeronautics, the Navy has had a definite policy with regard to its aircraft. Aircraft are required for two purposes which may be classified as Air Service and Air Force. Under Air Service, we consider that portion of Naval Aviation which is auxiliary to the fleet, such as scouting and control of gunfire. Under Air Force, we include that branch which is a striking force. This includes fighting, bombing, and torpedo planes based on carriers, as well as bombers and torpedo aircraft based on tenders. With a definite policy laid down in 1921, and a definite task before us which has been outlined down to the last detail as to the numbers and types required, we have been able to go ahead intelligently with a development program. Since the power plant is the heart of the airplane, this program has been predicated to a large extent on power plant development. The development is well underway and great strides have been made in bringing Naval Aviation and its important branches of Air Service and Air Force up to the requirements of the original program laid down.

NOTES.

NAVAL CONSTRUCTION IN 1925.

Important developments in the field of man-of-war construction took place during the past twelve months. Apart from the launching of noteworthy vessels at home and abroad, the year was memorable for the introduction of a long-term building program for the British Navy, having in view the systematic replacement of obsolete ships, as well as for the steady progress of warship construction in foreign countries. The volume of new combatant tonnage which went afloat during the year exceeded by a substantial margin that of any year since 1921, and approximated to 300,000 tons, more than half the total being represented by comparatively small vessels, ranging from cruisers to submarines. While it would certainly be premature to speak of a new competition in naval armaments, there are signs of a universal reawakening of interest in problems of sea power, which finds expression in a more energetic shipbuilding policy. Proof of this may be found, not only in the activity prevailing in the dockyards of the major Powers, but in the efforts which nearly every maritime State of secondary rank is making to augment its naval forces. Argentine, Brazil, Chile, Germany, the Netherlands, Soviet Russia and Turkey have one and all announced building programs which may or may not come to fruition. On the other hand, the economic causes which have done so much to delay the post-war renovation of navies are still in operation. Shipbuilding costs remain extremely high, and show no tendency to decline. Capital ships of the latest type cannot be built for less than £7,000,000 apiece. A cruiser of the standard 10,000-ton class costs about £2,500,000, or as much as a pre-war battleship. The modern destroyer is as expensive as a light cruiser was in 1914, and submarines have belied their former reputation for cheapness by costing from £350,000 to £950,000, according to tonnage. In these circumstances the creation of even a small navy of modern units must impose a severe strain on the exchequer of any State, and the wonder is, not that the present output of new fighting tonnage should be so much less than in pre-war years, but that it should be as large as it is.

BRITISH EMPIRE.

The battleships *Nelson* and *Rodney*, authorized in the Navy Estimates for 1922-23 and laid down simultaneously on December 28, 1922, were both launched, the former by Sir W. G. Armstrong, Whitworth and Co., at Walker-on-Tyne, on September 3, the latter by Cammell, Laird and Co., Ltd., at Birkenhead on December 17. Although nothing beyond the bare dimensions has been officially disclosed the general features of these ships are now fairly well known. It will, therefore, suffice to record such particulars as appear to be reliable, and for purposes of comparison the corresponding data of other battleships are given in the table.

In the absence of information as to the armor protection of the "Nelsons," it would be futile to extend the foregoing comparison. The figures given, however, leave no doubt that in the new vessels we shall possess the two most powerful battleships which have yet been designed. They are the last ships of capital rank which we are entitled to build until the year 1931, when two further vessels of 35,000 tons may be laid down.

Before proceeding to describe other ships of the year, reference must be made to the shipbuilding program which was announced in July. It provides

for the following new units up to the financial year 1929-30, inclusive:—In 1925-26: Four "A" class cruisers (10,000 tons), four gunboats, one floating dock. 1926-27: Two "A" class cruisers, one "B" class cruiser (8000 tons), six submarines, two depôt ships, four motor launches. 1927-28: One "A" cruiser, two "B" cruisers, nine destroyers, six submarines. 1928-29: One "A" cruiser, two "B" cruisers, nine destroyers, six submarines, one gunboat. 1929-30: One "A" cruiser, two "B" cruisers, one aircraft carrier, nine destroyers, five submarines, one fleet submarine, one net-laying ship. The total cost of this program is estimated at £58,000,000, of which £37,670,000 will fall on the Navy Estimates from the current year to 1929-30, inclusive. Large as these sums may be, they will be balanced in part by retrenchment in other directions which the methodical provision of new material will render possible. This aspect of the matter was explained by the First Sea Lord in his speech at the Lord Mayor's Banquet in November, in the course of which he said: "The Admiralty have during the past four or five years advocated continually the adoption of a steady program of replacement for naval defense, but owing to the instability of the various governments, who have never lasted more than a few months, no program has been accepted, and therefore no program adopted. Obviously, it is far more economical to have a steady program than to do as we have had to do in the last five years. The present Government, with its prospects of stability, has been able to accept a definite program, and so by stabilizing expenditure on naval construction it has enabled the Admiralty to exercise economies which would not have been possible under the previous circumstances. The effects are far-reaching, and include the demolition of many older ships now on the active list, with a corresponding reduction of expenditure on repairs, maintenance, and personnel."

Normal progress was made during the year on the five "County" class cruisers of the 1923-24 Estimates, all of which were laid down in the period September-November, 1924: the *Cornwall* at Devonport, the *Suffolk* at Portsmouth, the *Kent* at Chatham, the *Berwick* by the Fairfield Company at Govan, and the *Cumberland* by Vickers, Ltd., at Barrow-in-Furness. It is probable that all five ships will go afloat in the first half of this year. Two cruisers of the same design as the "Counties," to be named *Australia* and *Canberra*, are under construction by John Brown and Co., Clydebank, to the order of the Australian Government. Nothing is known about these seven ships save that they will displace 10,000 tons and carry 8-inch guns. Failing official data, we quote the following interesting comments on British cruiser design from the new edition of "Nauticus," the German naval yearbook: "It is probable that the 8000-ton 'B' ships will be a development of the 'Emerald' class, and the 10,000-ton 'A' ships an improvement on the 'Raleigh' class. Regarded purely as a fleet scouting cruiser, the *Emerald*, with her 7600 tons displacement, is unduly large. The improved 'Raleighs' would be useful as a support for the light scouting forces with the Fleet, now that no further fast battle-cruisers are to be built. Presumably, therefore, the new cruisers will represent an advance in speed in the 8000-ton class and an improvement in gun power in the 10,000-ton class. No change is likely to be made in the system of armor protection, which is common to both classes. We may thus expect to see an improved 'Emerald,' or 'B' class, with eight 6-inch guns in twin turrets, instead of the seven 6-inch on single mountings, and with a speed of 35 knots instead of 33. The larger 'A' type may have eight 8-inch guns in twin turrets and a speed of 31 to 32 knots."

Four additional "County" cruisers are to be laid down under the current Estimates, one at Portsmouth and Devonport, respectively, and the other two in private yards. Two new destroyers are under construction, the *Ambuscade* having been laid down by Yarrow and Co. on December 8, 1924, and the *Amazon* by Thornycroft and Co. in January, 1925. No par-

ticulars are available. A submarine, the *O 1*, of 1480 tons surface displacement, is on the stocks at Chatham, and two submarines for the Australian Government were ordered last year from Vickers, Ltd., to cost £300,000 each for hull and machinery. The large cruisers *Glorious* and *Courageous* are being rebuilt as aircraft carriers, and when completed may resemble the

Comparative Table of Battleships.

	Nelson class.	Royal Sovereign.	U.S.S. Colorado.	Japanese Mutsu.
Laid down	1922	1913–14	1919	1918
Length, feet	702	614	600	700
Beam, feet	106	88½*	97½	95
Draught, feet	30	28½	30½	30
Displacement, tons..	35,000	25,750	32,600	33,800
Speed, knots	23 (?)	23	21	23
Guns	9—16in., 50 cal., 12—6in.	8—15in., 42 cal., 14—6in.	8—16in., 45 cal., 12—5in.	8—16in. 45 cal., 20–5·5in.

* Original breadth, since increased by the addition of bulges.

Furious, of which they were originally sister ships. The cruisers *Effingham* and *Emerald*, which were completed during the year, have already been described in our columns. We regret that the Admiralty has not seen fit to release any details of their steam trials, though considerable interest attaches to the performance of the *Emerald* as the first 33-knot cruiser to be completed for the Royal Navy.

UNITED STATES.

Two aircraft carriers, *Saratoga* and *Lexington*, converted from battle-cruiser design, were launched during the year, the former on April 6 by the New York Shipbuilding Corporation, the latter on October 3 by the Fore River Shipbuilding Company. They are the largest vessels of any description so far built in the United States. In their modified form they have the following dimensions:—Length over all, 888 feet; extreme breadth, 106 feet; mean draught, 22 feet; normal displacement, 33,000 tons. Each is propelled by four sets of General Electric turbines, with electrical transmission, generating a total of 180,000 shaft horsepower, for a speed of 34.5 knots, steam being supplied to the turbines by sixteen Yarrow boilers in the *Saratoga* and the like number of White-Forster boilers in her consort. The propelling machinery is the most powerful which has yet been installed in any vessel. The arrangement of the superstructure generally resembles that of H.M.S. *Hermes*. A single funnel of immense size is placed at the extreme starboard side of the deck. Before it are the bridges, surmounted by a stump lattice mast, and two gun turrets. Abaft the funnel are two more turrets. By this arrangement all but a small part of the flight deck is left clear for aviation purposes, the deck having a total area of 5 acres. Protection is afforded by vertical and horizontal armor, and bulges are fitted. The armament is very powerful for ships of this type, consisting of eight 8-inch 55-caliber guns paired in turrets, and twelve 5-inch A.A. guns mounted on the broadside at the stern in two groups of six each. Each vessel is to carry seventy-two aeroplanes. The total cost is reckoned at over £9,000,000 per ship. Both will be completed before the end of the present year. Two of the eight 10,000-ton cruisers authorized during the last session of Congress were laid down on or about October 1. The design

is still confidential, but it is to be doubted whether these ships will carry the abnormal armament of twelve 8-inch guns with which they have been unofficially credited. A battery of this weight would be incompatible with the very high speed which is universally accepted as a *sine quâ non* in modern cruising ships, and would, moreover, probably involve the sacrifice of even the very light armor protection for which contemporary designs of the "Treaty" cruiser class are understood to provide. Submarine *V 3*, the last of a trio of large fleet boats, was launched on June 9. These are the heaviest submarines yet construction in America, and are intended to accompany the battle fleet on all occasions. Length overall, 341½ feet; breadth, 27½ feet; draught, 15½ feet; displacement, 2164 tons on the surface, 2520 tons submerged; Busch-Sulzer motors of 6500 brake horsepower, for a surface speed of 21 knots. Armament, one 5-inch gun on a high-angle mounting, six tubes and sixteen torpedoes. Great radius of action is a feature of the design. A still larger submarine, *V 4*, was begun on May 1st. It is stated that she will displace 3000 tons on the surface, *i.e.*, 220 tons more than the British *X 1*, at present the largest submarine in existence. The *V 4* is a mine-layer, but will also carry guns and torpedoes. Others of the class will be laid down when the necessary appropriations are made.

JAPAN.

Uninterrupted progress was made throughout the year with the building scheme authorized in 1922, the object of which is to provide the Japanese Fleet with a strong force of cruisers, destroyers and submarines, to compensate for its weakness in capital ships. The largest vessel to take the water was the aircraft carrier *Akagi*, which went afloat at Kure on April 22d. Laid down in 1920 as a battle-cruiser, she was among the ships condemned under the Washington Treaty, but was subsequently retained for conversion into an aircraft carrier in place of the *Amagi*, which was destroyed in the earthquake of September, 1923. The *Akagi* has the following dimensions, etc.:—Length, 763 feet; beam, 92 feet; draught, 21 feet; displacement, 26,900 tons; designed speed, 28.5 knots. She will be armed with ten 8-inch guns and sixteen 4.7-inch guns, most of the latter on high-angle mountings. Three light cruisers were launched during the year, the *Naka* on March 25th, the *Furutaka* on February 26th, and the *Kako* on April 10th. The *Naka*, a unit of the "Kuma" class, was delayed in construction owing to damage she sustained in the earthquake. She is the fourteenth and last of a series of ships which are virtually standard in dimensions, armament and speed, forming a uniform group of swift scouting cruisers which has no counterpart in any other navy. The displacement is 5500 to 5600 tons, the sea speed 33 knots, and the armament seven 5.5-inch guns. In the *Furutaka* and the *Kako* a much more powerful type of cruiser is represented, and one that is of special interest as reflecting Japanese ideas of what may be done without going to the limit of 10,000 tons allowed by the Treaty. The *Kako* and her three sisters—*Kinugasa* and *Aoba* are not yet launched—were planned in the first instance as 6000-ton ships, to be armed with 5.5-inch or 6-inch guns. After the Washington Conference, however, the design was recast on a basis of 7100 tons, and other radical changes were made. In the following table the *Kako* is compared with various foreign cruisers of approximately the same displacement:—

As will be seen from the following tabulation, the Japanese ship, though considerably the smallest of the four, has by far the heaviest broadside, and —on paper at least—is easily first in fighting power. So far as British naval requirements are concerned, it seems to us that a slightly larger version of the *Kako*, say, 8000 tons, with an armament of six 8-inch or 7.5-inch guns paired in turrets, would be preferable to an improved *Emerald* mount-

ing only 6-inch guns. If the *Kako* proves to be a steady gun platform, it would be impossible for any cruiser armed with 6-inch ordnance to approach her without running the risk of summary destruction, for in range, precision and smashing effect, the 8-inch gun is incomparably superior to the 6-inch. Nor need the *Kako* shrink from a duel with a "Treaty" cruiser of 10,000 tons. It remains to be seen, however, whether the mounting of this relatively tremendous armament in a ship of only 7100 tons has not involved some loss of seaworthiness and stability. At the same time, the reputation of Japanese naval constructors is a sufficient guarantee that the design has been carefully worked out in all details.

	I.J.N. Kako.	H.M.S. Emerald.	U.S.S. Omaha.	Spanish P.Alfonso.
Launched..	1925	1920	1920	1925
Length, feet	595	570	555	579
Breadth, feet	50¾	54½	55	53
Draught, feet	12	16½	14½	16½
Displacement, tons..	7100	7600	7500	7850
Speed, knots	33	33	33¾	33
Guns	6—8in., 4—3in.	7—6in., 3—4in.	12—6in., 4—3in.	8—6in., 4—4in.
Weight of broadside, lb. (main guns only)	1500	600	840	800

Two Japanese cruisers of the new "Treaty" class, *Nachi* and *Myoko*, which were begun in the latter part of 1924, are to be launched this year, and two sister ships, *Ashigara* and *Haguro*, will shortly be laid down. The type is 600 feet in length, 60 feet in breadth, and displaces 10,000 tons. A speed of 33.5 knots is specified. The main armament will be nine 8-inch guns in triple turrets, two forward and one aft. Several first-class destroyers of 1400 tons and 34 knots speed were launched in the course of the year; others are building. The construction of submarines was also pressed forward, and the year was notable for the completion of the large *I 51*, one of a group of boats designed for oversea service. They have a surface displacement exceeding 1500 tons, and a nominal cruising endurance of 12,000 miles. A supplementary shipbuilding program, comprising four 10,000-ton cruisers and many smaller craft, has been postponed for financial reasons, and new construction this year will be limited to a few destroyers.

FRANCE.

No news has yet been received of the trials of the three French 8000-ton cruisers—*Duguay-Trouin, Lamotte-Picquet* and *Primauguet*—all of which should have hoisted the pendant last year. Their performance on trial is awaited with special interest, as the machinery is designed for 100,000 shaft horsepower, a figure never before approached in a French man-of-war. Leading particulars of the class are as follows:—Length over all, 604 feet; beam, 56¼ feet; draught, 17¼ feet; normal displacement, 7880 tons. The armament consists of eight 6.1-inch guns in double turrets, four 3-inch A. A. guns and twelve torpedo tubes. Two aeroplanes and a catapult launching gear are provided. Two heavier cruisers are now under construction, the *Tourville* having been laid down at Lorient in August, 1924, and the *Duquesne* at Brest on October 30, 1924. The last-named vessel was launched on December 17th. They are 607 feet in length, 63 feet in breadth, displace 10,000 tons, and have machinery of 130,000 shaft horsepower. The main armament is to be eight 8-inch guns. A third and similar ship, *Suffren*, was begun at Brest in December, and a fourth and fifth are projected.

Both in the 8000-ton and the 10,000-ton class, armor has been virtually dispensed with in favor of the highest attainable speed. Several of the big 2362-ton flotilla leaders of the "Jaguar" class have recently made their trials, in every case with success. According to an official report of the Commission d'Essais, the *Tigre* maintained 35.93 knots, with a mean of 52,000 shaft horsepower for eight consecutive hours, in a disturbed sea and a stiff breeze. In the ninth hour a speed of 36.70 knots was reached with 58,000 shaft horsepower. The fuel consumption was remarkably low, and it is estimated that these boats have a radius of 4000 miles at economical speed. The reduction gearing for the *Tigre* and three of her consorts—*Jaguar, Chacal, Panthère*—was designed and manufactured by a British firm, the Power Plant Company, Ltd., of West Drayton, Middlesex, which has received gratifying reports of the behavior of the gears. The same firm, it is interesting to learn, is supplying the reduction gears for the cruisers *Duquesne* and *Tourville*, referred to above. The six flotilla leaders of the "Jaguar" class constitute a homogeneous light squadron of remarkable speed and power. Their sea speed may be taken as 35 knots, they carry five 5.1-inch guns and six torpedo tubes, and they are large enough to keep the sea in almost any weather. Fifteen more leaders, of a slightly heavier type, are to be built. Equally satisfactory were the trials of the 1430-ton destroyers, eighteen of which are already afloat. The nameship, *Simoun*, averaged 33.20 knots on an eight-hour run, and 34.30 knots for one hour during her trials in boisterous weather. Others of the class are said to have done still better. They are armed with four 5.1-inch guns and six tubes. Six further boats of this class were laid down in 1925.

Many French submarines were launched in the past year, including the *Narval, Dauphin*, and *Marsouin*, all of 1130 tons surface displacement and 16 knots speed. In all, nine units of this class are ready or approaching completion. Six larger boats—*Redoutable, Vengeur, Henri Poincaré, Poncelet, Pasteur*, and *Pascal*—with a displacement of 1550 tons and a speed of 18 knots are under construction. The first pair is to be equipped with two two-cycle Sulzer motors of 3000 brake horsepower. Twelve small submarines of the "Ariane" coast defense class, 600 tons and 14 knots, are being completed. In the present year it is proposed to lay down seven more submarines of the "Redoutable" group and two submarine minelayers. Although some delay has occurred in the completion of the cruisers, the French building program is, on the whole, maturing according to plan.

ITALY.

The cruisers *Trieste* and *Trento*, building at Trieste and Leghorn, respectively, have already been described in "The Engineer." Their displacement is 10,000 tons, and they are to have geared turbine installations of no less than 150,000 shaft horsepower. Sixteen new destroyers are building in various yards. They average 1250 tons displacement with 35.5 knots speed, and will be armed with four 4.7-inch guns and six tubes. Other construction in hand includes four submarines of 1360 tons and eight of 800 tons.

MINOR NAVIES.

The Spanish cruisers *Principe Alfonso* and *Almirante Cervera* were launched at Ferrol, the first on January 23d, the second on October 16th. Designed by Sir Philip Watts, of Armstrongs, they are ships of 7850 tons and 33 knots, armed with eight 6-inch guns. In form and dimensions they bear a close resemblance to our "Emerald" class.

Four destroyers of an uncommonly powerful type are in hand for the Netherlands Navy. They are being built in Holland to the designs and under the supervision of Yarrow and Co., Ltd., who are also supplying boiler material and certain other parts of the machinery. Particulars are

as follows:—Length, 322 feet over all; displacement, 1620 tons; Parsons geared turbines and Yarrow boilers for 31,000 shaft horsepower, and 36 knots. Armament, four 4.7-inch guns, two 3-inch A.A., six tubes. The Yarrow design was selected by the Dutch authorities out of a number submitted by firms in Great Britain, on the Continent, and in the United States.

Some further official details have reached us of the new German cruiser *Emden*, which was recently described and illustrated in our columns. The length on the water-line is 484 feet, extreme breadth 46¾ feet, normal displacement 6000 metric tons. Electric welding was extensively used in the construction of the hull, this being the first time, it is believed, that the process has been employed in a large man-of-war. The machinery consists of two sets of turbines with reduction gearing. The high-pressure turbines make 2435, the low-pressure turbines 1568, and the propellers 295 revolutions per minute at full power. There are four coal-burning boilers and six single-ended oil-fired boilers, housed in four compartments. At the time the ship was ordered it was deemed inadvisable to design her on an all-oil basis. She is expected to maintain a speed of 27½ knots for long periods under reasonable weather conditions. The eight 5.9-inch 50-caliber guns comprising the armament were taken from the reserve permitted by the Versailles Treaty. Twin mountings are now being manufactured, and when they are ready the ship will have her eight guns rearranged in pairs along the center line. A new system of fire control, embodying war experience, has been fitted. The foremast, which carries the gun director tower, has a diameter of no less than 60 inches. Davits have been eliminated, all the boats being lowered and hoisted by derrick booms pivoted on the foremast. The living quarters are superior to those of any preceding German warship, special attention having been given to this feature in view of the fact that the personnel of the German Navy is now on a long-service footing. A second cruiser of similar design is to be laid down this year at Wilhelmshaven, the Reichstag having voted the initial grant for the purpose. One new destroyer of 800 tons is on the stocks, and five more boats will shortly be laid down.—"The Engineer," January 1, 1926.

VESSELS UNDER CONSTRUCTION, UNITED STATES NAVY.

PROGRESS AS OF DECEMBER 31, 1925.

Type Number and Name	Contractor	Per Cent of Completion Jan. 1, 1926 Total	Jan. 1, 1926 on Ship	Dec. 1, 1925 Total	Dec. 1, 1925 on Ship	Contract date of Completion	Probable date of Completion
		AIRPLANE CARRIERS.					
Lexington	Beth. SB. Corp. (Fore River)	73.	70.4	72.	69.2		12/15/26
Saratoga	New York SB. Corp.	76.	75.3	74.6	73 9		11/ 1/26
		AUXILIARIES.					
Holland	Puget Sound Navy Yard	86.2	84.9	84 1	82.5		6/ 1/26
		FLEET SUBMARINES.					
V-3 (SF6)	Portsmouth Navy Yard	92.1	91.1	89.8	88.8		3/15/26
V-4 (SM1)	Portsmouth Navy Yard	24.1	22.1	21 8	19.8		4/ 1/27
V-5 (SC1)	Portsmouth Navy Yard	.....	.....	.....	.....		.....
V-6 (SC2)	Mare Island Navy Yard	.....	.....	.....	.....		.....

Authorized but not under construction or contract.

Authorized by Act of August 29, 1916—No funds as yet provided.

Destroyers (12) Nos. 348–359.
Transport (1) No. 2.
Fleet Submarines (3) Nos. 169–171.
Neff Submarine (1) No. 108.

Authorized by Act of December 18, 1924—Funds have been provided by the Act of March 4, 1925, for starting construction of two of the Light Cruisers and all six Gunboats.

Light Cruisers (8) Nos. 24-31.
River Gunboats (6) Nos. 43–48.

HYDROGEN AS A COOLING MEDIUM FOR ELECTRICAL MACHINERY.

At a meeting on Friday evening, October 23, of the New York Section of the American Institute of Electrical Engineers, Edgar Knowlton, Chester W. Rice and E. H. Freiburghouse presented a paper on Hydrogen as a Cooling Medium for Electrical Machinery, which was based on experiments made on a commercial machine. The authors gave the important characteristics of hydrogen when compared with air as a cooling medium for large high-speed electrical machines as: Lower density; higher thermal conductivity; higher forced heat convection; practically no damage to insulation by corona; prevention of fire.

Tests in hydrogen have shown a windage loss of 10 per cent of that in air. The total losses of large capacity alternators of 1800 or 3600 R.P.M. amount to approximately 2½ per cent, and the windage loss is one or more per cent of the rated capacity. By operating such a machine in hydrogen the windage loss is practically eliminated and results in about 1 per cent increase in efficiency.

HEAT CONDUCTIVITY.

Hydrogen has approximately seven times the heat conductivity of air or as high a thermal conductivity as the ordinary insulating materials. When a machine is operated in air, any spaces in the insulation or between the insulation and slot sides appreciably increases the thermal drop from the copper to the surface from which the heat is to be removed by the air. With hydrogen in the machine the gas spaces are no longer harmful in this respect, because of the greater heat conductivity of the hydrogen. The efficiency of heat removal by high-velocity gas blowing over the surfaces is approximately 1.3 times as great in commercial hydrogen as in air. In other words, the surface film temperature drop required in the hydrogen machine to transfer one watt per square inch is 77 per cent of that required for the same machine in air.

The potential gradient at which corona starts in hydrogen is approximately 60 per cent of that at which it starts in air. Thus a machine designed to operate in air without corona might produce severe corona when operated in hydrogen at atmospheric pressure. The destructive nature of corona in air on insulation is well known and it was, therefore, feared that insulation troubles would soon develop, if a standard machine without corona shields were operated in hydrogen. To test this point some preliminary comparative endurance tests were made in hydrogen and air on two similar samples of yellow varnished cloth.

The main point of interest was the strikingly different appearance of the corona in the two gases. In air there were vicious needlelike streamers coming out from spots along the wires, while in hydrogen the appearance of corona was soft and diffuse in character, and gave the impression that hydrogen corona would be harmless compared with air corona on the same sample and at the same voltage, as the tests proved to be the case.

In steam turbine generators there have been cases of fire starting on the surfaces of insulation without any electrical failure. If ventilated with air, the fire may spread with great rapidity, and injure the winding to the extent of requiring extensive replacement. Also a slight electrical failure, involving initially one or two coils, may start an equally serious fire. With hydrogen no fire can occur and the damage due to an electrical failure will be confined to a few coils.

FIRE PROTECTION OF GENERATORS.

Most of the large capacity air-cooled steam turbine generators are provided with pipes for the injection of steam or water in case of fire. The pipes are usually located near the ends of the windings, and when the steam or water is turned on, the ends of the windings are enveloped or covered by the extinguishing substance as the case may be. When there is a suspicion of a fire, the operator naturally wishes to be sure that the situation is serious before a machine is taken off the line and subjected to steam or water, and the tendency is to wait so long that damage may be done either by the fire or by the amount of steam or water that is turned into the machine. With a hydrogen filled machine no such disturbing condition can exist. The only failure to be considered is an electrical short-circuit or ground, and the well-known automatic devices now available should remove the machine from the line before any serious damage results.

A high-speed marine type alternator was selected for the tests, since it had fairly high windage losses, also because the outer shields which carried the armature were cast integral with the bearing supports, thus making a short closed-circuit system of ventilation readily obtainable.

DEVICES USED TO DETECT EXPLOSIVE MIXTURES.

Mechanical and electrical devices were used to detect explosive mixtures, automatically eliminate them, and prevent high pressures in case explosions should occur.

A small gasometer was connected to the point of lowest pressure in the machine, thereby maintaining a constant pressure slightly above that of the atmosphere, independent of temperature changes and leakage. Although leakage occurred on account of the crude joints between sheet iron and rough castings, the desired pressure was automatically maintained and contamination by air was prevented.

A recording device developed by C. Dansizen gave an accurate continuous log of the amount of impurity in the hydrogen. This device produced a graphic record of the difference in potential between the terminals of a pure metal filament placed inside the generator and through which a constant electrical current was automatically maintained. The resistance of the filament is a function of its temperature which depends upon the rate of heat transmission from the wire to the gaseous mixture, which, in turn, depends on the amount of impurity in the gas. With this apparatus a potential drop of 11.5 volts was observed with pure hydrogen and 12.5 volts with a mixture of hydrogen and one per cent of air impurity by volume.

Another safety device consisted of a metal filament, mounted upon a suitable support, placed within the generator and maintained at a temperature of from 700 degrees to 900 degrees C. by means of an electric current. A mixture of air and hydrogen, having less than 15 per cent of air, will com-

bine without explosion when brought in contact with the filament. The object of this device was to keep the air constantly being burned out of the hydrogen so that an explosive mixture could never be reached even though air should be continuously leaking into the machine.

RESULTS OF HEAT TESTS ON THE MACHINE.

Results of heat tests showed that the temperature rise was about the same when the machine was filled with hydrogen and supplying 133 per cent load as when filled with air and carrying 100 per cent.

The question most frequently asked the authors by those interested in this subject is in regard to explosions.

In order to answer this question some preliminary tests were made by E. W. Kellogg. The apparatus consisted of an iron pipe 4 inches in diameter by 24 inches long provided with a steam-engine indicator and spark plugs. The following maximum pressure rises were observed:

Gas Mixture by Volume	Maximum Pressure Lb. per Sq. In. Gauge
20 per cent H_2 in air	53
30 per cent H_2 in air	61
50 per cent H_2 in air	54
60 per cent H_2 in air	50
65 per cent H_2 in air	50

It was impossible to ignite mixtures containing more than about 65 per cent hydrogen. On the basis of heat content an ideal mixture of hydrogen and air should produce a temperature of 4100 degrees C. with a pressure of about 180 pounds per square inch gauge. That such pressures are not obtained in practice is due to dissociation and the cooling effect of the enclosure. To test the effect of an explosion on insulation, small pieces of double-cotton covered wire were placed in the bomb. Examination showed the outer layer of cotton to be scorched and slightly weakened. When the pressure was relieved by a paper vent over the end of the bomb, no scorching was detected. It may be concluded from these tests that an explosion in a machine would have no detrimental effect on the insulation. To determine the feasibility of limiting the explosive pressure in a machine to moderate values by the use of suitably placed vents, a model 5 feet by 3 feet 7½ inches by 2 inches inside was constructed. This represented as closely as convenient the various parts of the generator. The parts were full size except that the model was 2 inches thick. The various spaces simulated were: Fan space; space around ends of winding; air gap; armature core and ducts; chamber encircling armature core; intake chamber; connecting passage.

Pressures were measured by a steam-engine indicator started automatically just before the ignition of the gas. In general, the thicker the diaphragm the greater was the pressure developed by the explosion. As nearly as possible equal volumes of air and hydrogen were used but, in spite of the fact that the explosive range is considerably beyond this point in both directions, there were several times that the charge failed to explode, doubtless due to incomplete mixing. Twenty-five explosions were made and the highest recorded pressure (45 pounds per square inch) occurred when the venting diaphragms were replaced by steel plates.

It is the opinion of the authors that there is no more reason to fear the explosion of a machine filled with hydrogen than that of other apparatus

having considerable stored energy such as boilers, gas tanks, rotating elements, high-speed trains and automobiles, and various other familiar examples. Most of us fear a new condition if it has the possibility of accident, and accept as a matter of course the usual dangers of our daily lives.—"Power," November 7, 1925.

CENTRIFUGAL COMPRESSORS FOR DIESEL ENGINES.

By SANFORD A. MOSS.

Diesel engines are well established, as far as thermal efficiency is concerned, and most builders are now making efforts to secure a more favorable situation with regard to power output for a given weight and cost. One development in this direction, which is proceeding quite rapidly, is the use of high-speed centrifugal compressors for scavenging and supercharging both two-cycle and four-cycle engines. The use of high rotative speeds has decreased to a remarkable extent the weight and cost of many types of apparatus, and the use of centrifugal compressors gives some of this advantage to the Diesel engine.

The centrifugal compressor consists of an impeller carrying properly shaped blades, which rotates at high speed in a casing with suitable inlet and discharge conduits. There is appreciable clearance, so that no rubbing or sliding occurs. Such compressors are now in extensive commercial use, both in this country and abroad, for supplying air for various purposes. Types which supply air at pressures of from 1 to 5 pounds per square inch are being used commercially with Diesel engines.

The two-cycle Diesel engine, which has only an expansion stroke and a compression stroke, requires that air at a pressure of from 1 to 3 pounds per square inch be supplied by a scavenger of some type. This air is introduced into the cylinder through appropriate valves or ports when the piston is near the end of the expansion stroke, in such a way as to sweep out the exhaust gases through cylinder wall ports uncovered by the piston, and to fill the cylinder with a new charge.

Two-cycle scavenging is accomplished in four ways. The original "Crankcase Method" compressed the air in the crankcase by means of the lower side of the piston, but more recently there has been used a separate piston and cylinder operating in synchronism with the main piston.

The second or "Timed Method" uses a reciprocating blower not in exact unison with the power piston, but with such timing relative to it that the delivery of the air corresponds to the opening of the scavenging port.

The third or "Receiver Method" uses a reciprocating blower and a large receiver within which is an approximation to constant pressure. Claims have been made that a reciprocating blower with timing gives better performance than the constant pressure of a receiver, but it is nevertheless true that many engines with constant-pressure scavenging are as successful as those with timing. It is, however, difficult to secure a receiver large enough to give a really constant pressure with a reciprocating blower, and there is always an appreciable fluctuation of pressure. Hence, in order to secure proper scavenging the minimum pressure must have a required value.

The "Centrifugal-Compressor Method" of scavenging give a constant pressure. Tests show that the pressure actually needed for given results is lower than the average value of the irregular pressure of the Receiver Method.

Supercharging of four-cycle engines is desirable because of the fact that without it the charge of air at the beginning of the compression stroke has

a low density, due to a lower pressure and a higher temperature than that of the atmosphere. Supercharging gives an increase in the amount of charge, and hence in the amount of power delivered, by supplying the intake manifold with air at a pressure appreciably above atmospheric.

By increasing the pressure of the scavenger of a two-cycle engine, any desired amount of supercharging may also be obtained, provided that the air ports remain open after the exhaust ports have closed or that the engine exhausts against a back pressure.

INSTALLATIONS OF CENTRIFUGAL COMPRESSORS FOR SCAVENGING AND SUPERCHARGING.

A great many installations of centrifugal compressors have been made by the Brown-Boveri Company, of Switzerland, C. H. Jaeger, of Leipzig, the British Thomson-Houston Company, and other European builders. Most of these are driven by high-speed direct-current motors and are for marine use.

The most outstanding application of a two-cycle engine with a centrifugal scavenger is on the *Aorangi*, a quadruple-screw liner of 23,000 tons displacement. Four 6-cylinder, two-cycle Sulzer engines developing a total of 13,000 shaft horsepower at 127 R.P.M. give the greatest power ever used in a Diesel ship.

The scavenging and supercharging air is supplied by two centrifugal blowers, each capable of delivering 34,000 cubic feet of air per minute against a pressure of 1.75-2.1 pounds per square inch, while a third unit acts as a stand-by. Besides the usual scavenging ports uncovered by the stroke of the piston, there is a row of automatic multiple-disk valves just above them, which open when the exhaust pressure falls below that of the scavenging air and which remain open after the exhaust valves have been closed on the return stroke, until the air pressure in the cylinder equals that of the scavenging air. This provides for greater than atmospheric pressure in the cylinder at the beginning of compression, or true supercharging.

Another application in an entirely different field is that of the new Baldwin locomotive, whose driving element is a Knudsen Diesel engine scavenged and supercharged by a General Electric Company centrifugal compressor.

The Knudsen Diesel engine is of the two-cycle, inverted V-type, corresponding to an opposed-piston engine with the cylinder divided in the middle to form an acute angle so that the pistons are operated from separate crankshafts which are geared together to maintain the required synchronism and to provide a common driving shaft. The compressor supplies air at from 1½ to 3 pounds per square inch, whereby scavenging and supercharging from 10 per cent to 30 per cent are obtained, according to the speed and power requirements.

In England a number of installations have been made with centrifugal compressors for two-cycle scavenging driven by steam turbines.

The greatest use of centrifugal compressors for supercharging four-cycle engines has been in Germany, and they are now used there on practically all new projects.

The supercharged four-cycle Diesel engine which has been in operation for the longest time is that of the Norwegian ship *Tiradentes*. When the ship was first operated it did not develop sufficient speed, and a supercharger was added which increased the indicated horsepower from 3100 to 3600 and the speed of the ship from 10½ knots to 11½ knots.

The engines were built by the Allgeméine Elektricitäts Gesellschaft, of Germany, under license from Burmeister & Wain. In order to avoid increase of compression pressure with supercharging, the clearance space was increased by lowering the position of the pistons about 0.2 inch.

The supercharger is a centrifugal compressor driven by a direct-current motor, built by the Brown-Boveri Company and similar to their scavenging compressors already described. The supercharging pressure is only 0.85 pound per square inch, which requires an electrical input of 33 kw.

With the supercharger in operation, the exhaust temperature varies from 806 degrees to 842 degrees, which is higher than with ordinary four-cycle engines. The exhaust is entirely invisible and there are no soot deposits in the exhaust conduit. Every 40 days the exhaust valves have to be reground instead of every 60 days as would ordinarily be the case, and a somewhat increased amount of cooling water is required. These points are regarded as a very small price to pay for the increased power.

Since the *Tiradentes* installation a number of other four-cycle Diesel-engined ships have been supercharged in practically the same way. Among these may be mentioned the *Tampa, Tungsha, Taiwan,* and *Tortugas,* belonging to the Wilhelmson Company as does the *Tiradentes.* Some of these vessels frequently visit the United States. Two 14,000-ton Diesel passenger liners have recently been completed in Germany, with superchargers as an integral part of the design. These are the *Monte Sarmiento,* and the *Monte Olivia.* A number of other supercharged ships are also in the process of construction or have just been completed.

Although the matter of supercharging Diesel engines has only recently begun to attract attention, the proposition is really quite old and many of the early workers on both Otto-cycle and Diesel-cycle engines mention it. Propositions were also made years ago to use separate blowers with Diesel engines in mountainous regions, in order to bring the power to the value obtained near sea level.

CENTRIFUGAL COMPRESSORS FOR DIESEL ENGINES.

The Centrifugal Compressor resembles in a general way the well-known fan blower. However, the details of design of the conduits in the casing leading to and from the impeller, and of the vanes on the impeller, are attended to more carefully and the rotative speeds are much higher, varying from 3000 to 10,000 R.P.M. The speed of the air through the apparatus is also comparatively high. For these reasons the vanes and passages must be designed to handle the air without shock. The air leaving the impeller possesses a velocity of the order of magnitude of the peripheral speed of the impeller, which represents an amount of energy equal to about one-half the power output of the driver. There is a collecting passage or diffuser beyond the impeller which receives this air and decreases its velocity without shock, so that the energy represented by the velocity is converted into energy represented by pressure. Due to the high speed and general design, centrifugal compressors are comparatively light and compact.

Centrifugal compressors have been in competition with older types of compressors for many years and their use is rapidly increasing. Many actual comparisons have been made of efficiencies of the several types, and these have shown that a centrifugal compressor and good compressors of the older types have efficiencies which are essentially equal when both are in good condition. A centrifugal compressor is subject to no deterioration under usual circumstances, and hence maintains its original efficiency. The older compressors always involve rubbing or sliding parts. In some cases these are maintained at appreciable expense so as to retain the initial efficiency. With average maintenance there is soon enough leakage to materially decrease the efficiency.

Drivers for centrifugal compressors for Diesel engines are often direct-current motors. In case of installations on board ship this is usually desirable for the reason that there is an electric generator driven by an auxiliary engine which supplies power for winches and other auxiliary machinery in

port, and which therefore gives capacity for supplying compressor motors at sea. The European sets have usually been directly connected to especially designed high-speed motors running at from 2500 to 4500 R.P.M. The American sets have usually used electric motors with lower speeds and centrifugal compressors with higher speeds, with gearing between. This has given the possibility of selecting the speed of each unit in the most advantageous way and without compromise for the benefit of the other. The low-speed electric motor avoids difficulty of commutating, etc., and the high-speed compressor makes cost, weight, and space a minimum.

In some cases, as in the Knudsen locomotive previously described, a centrifugal compressor for supercharging or scavenging is directly geared to the engine. In such a case, and if the engine conditions are such as to give a proper volumetric efficiency, there is automatic pressure compensation for the variation of friction through the passages with engine speed, and if there is supercharging in addition, this automatically varies as the square of the engine speed. So far as is known, this is the only use thus far made of a geared supercharger on a Diesel engine. There have been made, however, a few installations of geared superchargers on Otto-cycle engines for racing automobiles by the author and others. A great many experiments on geared superchargers for Otto-cycle airplane engines have been made, but no commercial use has occurred as yet.

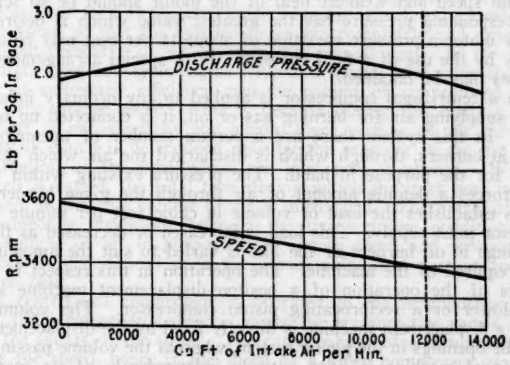


FIG. 1. DISCHARGE PRESSURES AND SPEEDS OF GENERAL ELECTRIC CENTRIFUGAL COMPRESSORS FOR FORD ORE VESSELS.

(Based on air at 14.4 lb. per sq. in. abs. and 60 deg. fahr.)

In Great Britain certain maritime rules require that passenger ships have a number of steam-driven pumps and steam for heating the ship, etc. Hence most British ships with centrifugal compressors for Diesel engines have had steam turbines as drivers.

There have not been enough installations of separate blowers for stationary Diesel engines to indicate the type of driver which would be most usual. It would be possible to drive the compressors direct by alternating-current induction motors at 3600 R.P.M. synchronous speed. The General Electric Company is now supplying a great many such alternating-current sets for miscellaneous industrial purposes.

The use of an exhaust-gas turbine is also possible with both Diesel-engine scavenging and supercharging, as discussed later.

Centrifugal-Compressor Characteristics. The characteristics of a centrifugal compressor are so different from those of a positive-displacement blower that consideration of them is worth while. The upper curve of Figure 1 gives the pressure produced by a centrifugal compressor when operated at various volumes. As will be seen, there is a comparatively slight variation of pressure as the volume varies. This is the curve actually produced by the Ford compressors. This curve is for a constant position of the field rheostat of the direct-current motor which drives the set. All General Electric Company compressors give similar curves whether driven by direct- or alternating-current motors or by steam-turbine sets.

In the case of a direct-current motor there is usually a field rheostat which can be set at various positions so as to give speed variation. With any one setting of this field rheostat the speed decreases somewhat as the load increases, as shown by the lower curve of Figure 1.

With the field rheostat in one extreme position, so that there is the "full field" on the motor with no rheostat resistance in series with the field coils, there is obtained the minimum speed and lowest pressure. This should be so selected as to be the value required during starting or low-speed operation of the engine. By turning the field rheostat by hand, resistance is added to the field circuit and the field is "weakened," which increases the speed. The maximum speed and weakest field of the motor should be so selected that the corresponding pressure has the greatest value which is desired. In an ordinary motor a pressure variation of about 15 per cent may be obtained in this way by the use of a field rheostat. With special arrangements, greater variations may be obtained.

When a centrifugal compressor is applied to any ordinary industrial use, such as supplying air for burning gas or oil, it is connected up to a piping system. In this system there are a certain number of openings, such as nozzles in burners, through which is discharged the air which the machine supplies for the purpose in hand. The pressure existing within the piping system forces a definite amount of air through the given burner openings and thus establishes the load or volume in cubic feet per minute which the compressor must supply. This load is increased or decreased as the number of openings in oil burners or the like is varied to suit the amount of work that is required of the machine. The operation in this respect is the direct antithesis of the operation of a positive-displacement machine such as a Roots blower or a reciprocating piston compressor. The volume passing through a displacement machine is directly fixed by the displacement regardless of the openings in the piping system, whereas the volume passing through a centrifugal machine is fixed entirely independently of its speed by the number of openings which are arbitrarily introduced into the piping system. In other words, the centrifugal compressor gives a practically constant pressure while automatically adjusting itself to the volume demanded by the number of burners or other openings required at the time.

While the pressure produced by the machine is nearly constant, regardless of the volume, this is only true within certain limits. The exact pressure when a given volume is made to flow through the compressor by a given number of openings in burners or the like, is given by a characteristic curve such as in Figure 1. For very large volumes the pressure is somewhat below that at the rated volume. However, if a machine is properly selected for the work in hand, whether for a Diesel engine or for oil burning, the pressure will be practically constant for all ordinary variations of volume. Unfortunately, however, cases sometimes arise where machines are not properly selected. The volume imposed on a machine by the work in hand may then be very much less than that for which the machine is suited, in

which case there are the pulsations alluded to later; or the volume may be much greater, in which case the pressure delivered by the machine is low and the power is high.

When a centrifugal compressor of such characteristics is applied to scavenging or supercharging a two-cycle or four-cycle engine, the operation is essentially as described above if the valve and port arrangements and engine speed are such that enough air flows into the cylinder at each charge to bring the pressure within the cylinder up to a value approaching that in the discharge pipe of the compressor. Under such circumstances the cylinder displacement is practically filled at each stroke, and the total load on the compressor is directly given by the total displacement of the engine.

It must be understood that the engine displacement is never quite filled with air at atmospheric temperature and at the pressure existing within the compressor discharge pipe. The temperature within the cylinder is somewhat higher and the pressure somewhat lower than these values. Hence the actual volume of air is 80 to 90 per cent of the volume given by the direct displacement, and the ratio is called "volumetric efficiency." Values of this factor have been determined for some Otto-cycle engines and range from 85 to 88 per cent. Values for Diesel engines are at rated speed undoubtedly nearly the same if the speed and valve and port arrangements are proper.

The above statements apply directly to a four-cycle engine or to a two-cycle engine with a special inlet valve which remains open after the exhaust valve is closed, or to a two-cycle engine running at a speed exactly suited to the port areas, so that the exhaust gases are completely and exactly displaced without discharge of air through the exhaust port. In the extreme case of a high-speed two-cycle engine without a special inlet valve there is no such thing as volumetric efficiency, and the volume of air required is independent of the engine speed. This is because during the entire time that the inlet port is uncovered by the piston, there is flow of air through it, the amount of which is determined only by the pressure in the air conduit and the fraction of the time which the inlet port is opened. When the engine speed is varied, the total time during which the inlet port is opened is unchanged, so that the volume of air is unchanged. There are, of course, many intermediate conditions between the two extreme conditions above specified—volume of air in proportion to engine speed and independent of engine speed—which depend upon the actual valve and port arrangements in any given case.

When a centrifugal compressor is operated in the neighborhood of its rated load, if the compressor speed is varied the pressure rise varies approximately as the square of the speed. The rated volume of a compressor varies directly as the speed, hence if the speed of a given compressor is doubled, the volume to which it is suited is also doubled and the pressure rise which it produces at the double volume is quadrupled. A centrifugal compressor geared to an engine is therefore automatically maintained at its rated volume, since as its speed is increased the volume demanded by the engine is increased in the same ratio. Friction losses through valves and ports vary directly with the square of the volume; hence, as already stated, a geared compressor automatically gives a pressure rise which overcomes friction losses at all speeds, and, if there is supercharging, gives supercharging pressure varying with the square of the speed.

A centrifugal compressor serving a Diesel engine should always have a butterfly or other type of gate valve at the compressor inlet. The compressor is started with the inlet valve closed before the engine is started, and operated at the lowest speed possible by using the field rheostat or the like. The Diesel engine is then started, and as it attains speed the compressor inlet valve is gradually opened. The valve is opened completely if the full pressure of the compressor is desired, or partially if the full pressure is not

desired. After the inlet valve is opened completely with the compressor at the lowest speed possible, the speed is increased by means of the field rheostat if necessary. When three centrifugal compressors are arranged so as to serve two engines, additional butterfly or other gate valves are desirable at discharge as well as at inlet.

As will be seen from the characteristic curve of Figure 1, when the load on a centrifugal compressor is very much below normal, the pressure rises with increase of load. This gives an unstable characteristic and therefore a centrifugal compressor operating at a load very much below normal or at no load such as when the compressor is running with discharge gate open before the Diesel engine is started, undergoes so-called "pulsations" or pressure fluctuations accompanied by a back rush of air out of the inlet and a peculiar noise. This is eliminated by throttling or completely closing a valve at the inlet, and hence centrifugal compressors are always controlled by such a valve during starting or light-load operation as already specified.

Every Diesel installation has some sort of a high-pressure air compressor for starting and maneuvering purposes, and many have air compressors for the injection of fuel. Recent installations of scavengers or superchargers have been so arranged that they supply these compressors also. This makes it possible to use a smaller compressor for a given amount of air or gives a larger amount of air from the same compressor. A scavenger for 2-pound pressure will increase the compressor capacity about 15 per cent. Centrifugal superchargers and scavengers are always provided with inlet conduits so that air supply can be drawn from outside the engine room. It has been proposed to connect these conduits to parts of the ship which need to be ventilated so as to take the place of ventilating blowers which would otherwise be required.

ADVANTAGES OF CENTRIFUGAL COMPRESSORS FOR TWO-CYCLE SCAVENGING.

Weights and Dimensions of Reciprocating Blowers and Centrifugal Compressors. The most important advantage of scavenging a two-cycle engine by a separate centrifugal compressor is the great reduction of cost and weight. See Figure 2. When a Diesel engine is designed for a separate air supply, besides dispensing with the huge blowing cylinder with all its accessory valves, rods, etc., the massive base can be made considerably shorter and more than proportionately lighter.

A comparison of the weights and dimensions of the Ford Diesels, which have separate blowers, with the engines of the ship *Challenger* which are of similar design but with reciprocating scavengers, was made by the Sun Shipbuilding and Dry Dock Company of Chester, Pa., who built both engines. This shows that the weight has been reduced from 275 pounds per brake horsepower to 246 pounds per brake horsepower, including the spare compressor set, which is a net saving of 36,000 pounds or 4.4 per cent. The length was also calculated to have been reduced 8 feet or 13 per cent. The use of old patterns in the Ford engines did not permit of the full reduction possible with the centrifugal-compressor system.

Improvement in Low-Speed Operation has been obtained with centrifugal scavengers due to increased supply of scavenging air. This is due to use of full pressure with decreased air friction resulting from the low speed. Reports from abroad and from the two installations in the United States already alluded to, uniformly praise centrifugal-compressor scavengers for giving ease in maneuvering. The starting of a Diesel engine is also facilitated since the centrifugal compressor, being started in advance, gives a full supply of warm air for the first firing stroke.

Efficiency. No data are available as to the actual efficiency of reciprocating scavengers, but efficiencies greater than 75 per cent are actually guaranteed for centrifugal compressors. The efficiency of a reciprocating blower is

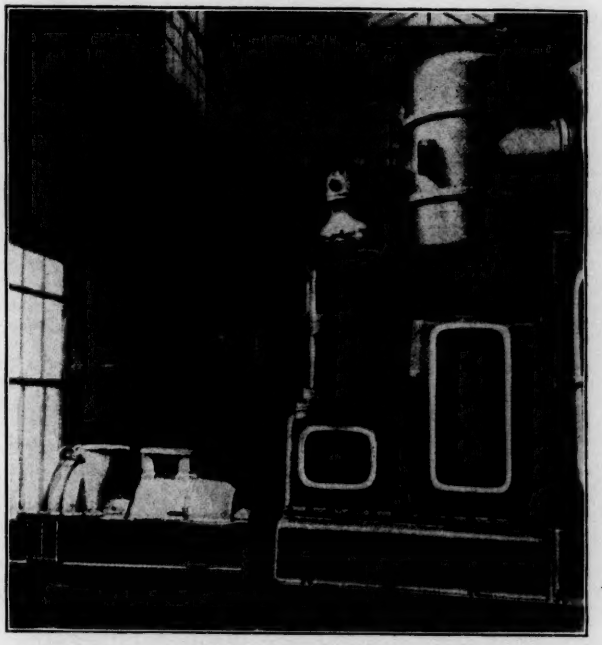


FIG. 2. VIEW SHOWING COMPARATIVE SIZE OF A TWO-CYCLE RECIPROCATING SCAVENGING BLOWER AND A CENTRIFUGAL COMPRESSOR OF THE SAME CAPACITY.

probably lower on account of the losses in volumetric efficiency and of friction due to driving a low-pressure blowing piston from the heavy crankshaft of a Diesel engine. However, it is customary to assume that such losses are about equal to the losses in the electric generator and in the motor and the centrifugal compressor. On the basis of this assumption the relative power taken from the crankshaft is directly proportional to the pressure rise. It has already been pointed out that the pressure rise necessary to secure a given quantity of scavenging is somewhat less with a centrifugal compressor than with a reciprocating blower, so that the power would be reduced in practically the same proportion. This conservative estimate shows the power for scavenging a centrifugal compressor to be about 70 per cent of that for a reciprocating blower.

Incidental Advantages. If an accident happens to a reciprocating scavenger the entire plant is put out of commission, which, if it should occur to the main engine at sea, would make the vessel helpless until repaired. On account of the low cost, most installations of centrifugal compressors are made in duplicate. This gives an insurance against accident which is not possible with reciprocating scavengers.

The valves and piston rings of a reciprocating scavenger must, of course, be carefully kept in order to keep up the initial scavenging pressure and to maintain the engine power. This maintenance is entirely eliminated with a centrifugal scavenger. A margin in the amount of scavenging air is provided for with but little additional cost in the case of a centrifugal scavenger. The size and weight of a reciprocating scavenger make builders loath to provide such a margin with it. The extra scavenging air thus possible gives cleaner exhaust and less deposits in cylinder and exhaust conduit, as well as a possibility of a slight increase of power.

SIZE OF CENTRIFUGAL COMPRESSORS FOR TWO-CYCLE SCAVENGING.

The displacement of a reciprocating scavenger is usually selected on the basis of an increase of about 25 per cent over the displacement of the main engine. This gives no information as to the volume required of a centrifugal scavenger because of the great uncertainty regarding the air actually delivered by a reciprocating machine. Some foreign builders of centrifugal scavengers have found it desirable to use diffusers which can be adjusted to suit the volume actually required when the apparatus is installed in connection with the engine. Approximate computations of the air required from the centrifugal compressor are as follows: The displacement of the reciprocating blower of the ship *Challenger* corresponds to 4 cubic feet per horsepower. A volumetric efficiency of about 87.5 per cent gives a round figure of 3.5 cubic feet of actual air per brake horsepower required from a centrifugal scavenger, and this figure has been used by some builders. Computations made on the basis of the displacement of the main engine are somewhat uncertain owing to uncertainty as to the exact amount of exhaust gas scavenged out. This may be less than the displacement, equal to the displacement, equal to the displacement plus the clearance volume, or more than this so as to force some air through the exhaust port. The flat pressure curve of the centrifugal compressor gives little difficulty as far as delivery of air is concerned, and the compressor operates well even though it is somewhat too small or too large. Too large a motor as driver also gives no difficulty, which leaves the possibility of an overload on the motor as a principal item to be considered. So much depends upon the engine design and the scavenging pressure as to make the selection of the exact size of the centrifugal compressor more or less of an empirical matter, to be based on experience which has been obtained with each engine design.

SUPERCHARGING EXPERIMENTS.

A great many experiments have been made in which two-cycle and four-cycle engines have been supercharged, and an appreciable gain in power found.

These experiments involved supercharging to two quite different degrees. In some the amount of supercharging has been comparatively slight so as to mainly overcome the friction drop through inlet conduits and valves. In others a much greater amount of supercharging has been used. The experiments have been mainly with four-cycle engines.

A slight amount of supercharging with blowers delivering air at a pressure of 400 to 600 millimeters of water, or 0.75 pound per square inch has been found to give increases of power much greater than proportional to increased charge in the displaced volume, the reasons for which are discussed later. This type of supercharging is the only one which has come into actual commercial use at the present time. A large amount of supercharging varying from 1.5 pounds per square inch to 30 pounds per square inch has been used in other experiments and the net power obtained actually, or computed after allowing for power for driving the supercharger. In all cases there was a gain in this net power, even though it involved an increase in the pressure at the end of expansion. A number of experiments have been published by Rateau and by the author on the supercharging of an Otto-cycle engine by means of an exhaust-gas turbine for use on airplanes operating at high altitude, but while the use of such an exhaust-gas turbine has often been proposed for a four-cycle Diesel engine, no experiments have ever been published. It is understood that a number of tests were made on a European locomotive some years ago. A great many airplanes with four-cycle Otto

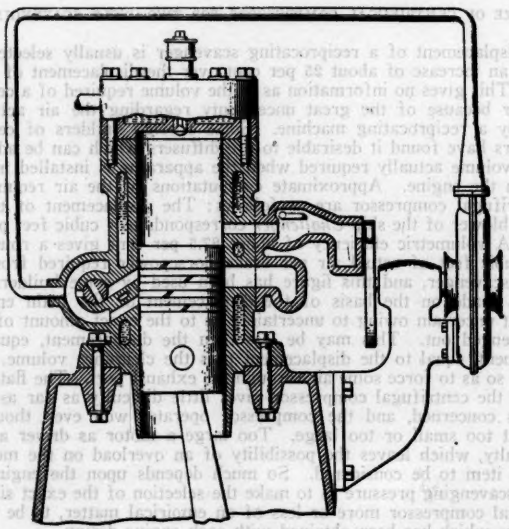


FIG. 3. SCAVENGING A TWO-CYCLE DIESEL ENGINE WITH AN EXHAUST-GAS TURBINE.

engines and exhaust-gas turbines, have been supplied for commercial use by Rateau in France, and by the General Electric Company in this country. The Brown-Boveri Company have advertised that they will supply similar exhaust-gas turbines for Diesel engines. So far as is known, however, no commercial installation of an exhaust-gas turbine on a Diesel engine has ever been made.

Some unpublished experiments have recently been made in the United States, in which an opposed-piston two-cycle engine has been supercharged with initial pressures from 1½ to 30 pounds per square inch and appreciable gains in power and savings in fuel consumption obtained.

A few unpublished experiments have been made with an exhaust-gas turbine supercharging a two-cycle engine according to a plan devised by the author and shown in Figure 3; this is explained later in the paper.

THEORY OF SUPERCHARGING A DIESEL ENGINE.

As already explained, supercharging involves forcing air into the cylinder by a compressor or blower of some kind, so that at the beginning of compression there exists within the cylinder an absolute pressure greater than atmospheric. Without supercharging, the pressure would be below atmospheric for four-cycle engines. In the case of two-cycle engines, supercharging means increase of the pressure of scavenging air so that the absolute pressure in the cylinder at the beginning of compression is greater than the value slightly above atmosphere which ordinarily serves to eject the exhaust gases. Of course, the two-cycle engine can only be supercharged if there is some valve or port arrangement which permits of adding additional charge after the exhaust ports are closed.

Such supercharging changes the power delivered by the Diesel engine in the following ways which are later considered in detail.

In the first place, if the inlet and exhaust valves have a proper lap, the supercharging pressure can result in scavenging the clearance volume of exhaust gases so that the air charge is greater than the displacement due to the filling of the clearance volume with air at the supercharging pressure.

In the second place, the removal of the exhaust gases gives a cleaner charge, particularly in the vicinity of the fuel injection nozzle, which has been found to permit of the burning of a greater amount of oil—that is, a decreased air excess.

In the third place, the increase of power with a given water-jacketed surface can decrease the percentage of heat carried away by the cooling water, which increases both power and thermal efficiency.

In the fourth place, the production of an increased amount of power with given cylinder and crank friction increases the mechanical efficiency.

In the fifth place, the expansion of an increased charge to a given final volume gives a decrease in the area of the theoretical indicator card.

The relative effect of each of these items, of course, varies with different engines.

Clearance and Compression. The amounts of supercharging ordinarily contemplated, say, from 1 pound per square inch to 5 pounds per square inch above the absolute pressure of the atmosphere, involve an increase of the amount of charge of from 6 to 30 per cent. If the clearance volume with supercharging is maintained at the same percentage as without, there will result an increase of compression pressure, and this has occurred in some of the cases previously mentioned. In most cases, however, the clearance volume has been increased in the same ratio as the absolute inlet pressure, so as to maintain the original compression pressure.

Scavenged Clearance Volume. Whether there is supercharging or not, the inlet valve always opens just before the piston reaches the end of the exhaust stroke, in order to properly start the admission of the new charge.

Whether there is supercharging or not, the exhaust valve always closes somewhat after the end of the exhaust stroke and when the piston has started on the suction stroke. This gives a period when the piston is near the top of the stroke at the end of the exhaust and the beginning of the admission, when both inlet and exhaust valves are open together. When a supercharger is used, there is a pressure in the inlet manifold appreciably higher than that in the exhaust manifold, and hence the new charge will start to enter the cylinder even during the latter part of the exhaust stroke. Therefore some of the exhaust gases in the clearance space, which ordinarily are never pushed out by piston displacement, will be forced out by the supercharging pressure.

If the valve openings are properly retimed to suit supercharging, it is quite possible that the clearance space may be filled with fresh charge and the exhaust gases entirely removed without discharge of air through the exhaust port. As the piston proceeds on the suction stroke with the exhaust valve fully closed and the inlet valve fully open, the cylinder (including the clearance space) will then be filled with a fresh charge at a pressure below the pressure of the supercharger by the amount of the valve and port friction. The increase of charge due to supercharging is due to filling the cylinder at this increased pressure as well as to replacing the exhaust gas in the clearance space.

If the original compression pressure is maintained, the clearance volume must be increased in the ratio of the absolute pressures at the beginning of the compression stroke with and without supercharging. Hence the clearance space which is filled with the added fresh charge in the case of supercharging is larger even than the clearance space in the original engine.

Curves are given in Figure 4 showing the percentage increase in amount of new charge due to all of these facts, with pressures of ½, 1 and 2 pounds per square inch below atmospheric pressure in the cylinder at the beginning of the compression stroke and various amounts of supercharging.

The increase in power with given supercharging depends not only on the theoretical increase in charge as given by the curve, but also on the amount of fuel burned with a given amount of charge, which depends upon considerations given beyond.

Improvement of Combustion Due to Supercharging. Some of the experiments previously cited have given an increase of power with Diesel engines greater than that theoretically computed by the principles of the preceding paragraphs, due to the fact that it has been found possible to burn greater percentages of oil. As is well known, the charge of oil injected into the air of a Diesel engine requires about half of the air for theoretical combustion. The amount of oil injected at maximum power is always made as great as is consistent with a clean exhaust. The absence of exhaust gases from the clearance volume facilitates the chemical act of combustion and the rapidity with which it spreads. The increased pressure with which the supercharged air enters the cylinder through the inlet valve also creates currents so that there is certain to be pure air in the vicinity of the fuel-injection nozzle. Without supercharging there is, of course, a pressure in the exhaust manifold higher than in the inlet manifold, so that during the valve lap previously referred to, exhaust gases may actually enter the inlet manifold and be returned again into the cylinder near the fuel-injection nozzle when the exhaust valve closes. All of these actions explain the well-verified fact that an increased percentage of fuel can be burned in a Diesel engine when supercharging is used, giving a decreased air excess. Hence there has been the great increase in engine power obtained with a small amount of supercharging, 400 to 600 millimeters of water, used in many recent German installations.

Water-Jacket Loss. If in an unsupercharged and a supercharged Diesel engine the same compression pressure and the same percentage of air charge

are required for the combustion of the oil, there will be an increased amount of heat liberated with supercharging without any change in temperature. The surface of the combustion chamber in the two cases will be nearly the same, and hence there will be the same amount of heat lost to the jacket water during the early part of the stroke. The jacket-water loss will then be a smaller fraction of the total heat of combustion, which will result in a slight improvement in efficiency. If, however, the percentage of air which

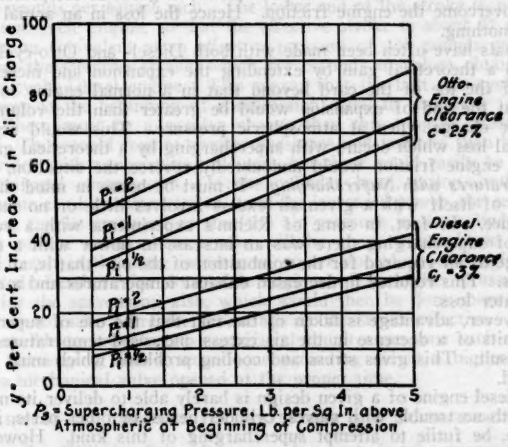


FIG. 4. THEORETICAL INCREASE IN AIR CHARGE DUE TO SUPERCHARGING. (Pi = initial pressure without supercharging in cylinder at beginning of compression, pound per square inch below atmospheric.)

is used to burn the fuel is increased and the air excess diminished, there will of course be an increase of temperature and an increase of the jacket-water loss. However, the considerations previously mentioned will tend to keep this a small percentage for the supercharged case. It is true that the combustion in a supercharged engine will last through a somewhat greater portion of the stroke than otherwise, and the temperature when the exhaust valve opens will be somewhat higher. This will tend to increase the jacket-water loss during the latter part of the stroke where it is comparatively small. On the whole, however, it is to be expected that supercharging will decrease the relative amount of heat loss to the jacket.

Increase in Mechanical Efficiency. The increase of power from a given cylinder due both to the increased fuel ratio and the increased charge, decreases the relative percentage of friction loss. This is a major item and is the primary reason for the observed increase in thermal efficiency which has always accompanied supercharging. The theoretical loss due to decreased ratio of expansion mentioned in the next paragraph diminishes this gain somewhat, and the decreased jacket-water loss mentioned above increases it somewhat, but both of these are probably minor items.

Incomplete Expansion. In an engine without supercharging the volume of the gas at the end of expansion is equal to the volume occupied by the initial air charge when at a pressure slightly below atmospheric pressure, both of these volumes being equal to the cylinder displacement. With a

supercharged engine the volume of the gases at the end of expansion is equal to the cylinder displacement as above, but this volume is now equal to the volume of the initial air charge when at a pressure appreciably above atmospheric pressure. Hence a supercharged engine does not expand the gases to as low a pressure as in the normal case. In other words, there is a small area at the toe of the theoretical indicator card which is lost. The indicated pressure in the cylinder is comparatively small at this end of the stroke and is but little greater or perhaps not as great as the pressure necessary to overcome the engine friction. Hence the loss in an actual engine is little or nothing.

Proposals have often been made with both Diesel- and Otto-cycle engines to obtain a theoretical gain by extending the expansion and increasing the length of the toe of the card beyond that in a normal engine so that the volume at the end of expansion would be greater than the volume of the initial air charge when at atmospheric pressure. This would replace the theoretical loss which occurs with supercharging by a theoretical gain. The effect of engine friction would undoubtedly reverse the situation, however.

Temperatures with Supercharging. It must be borne in mind that supercharging of itself with a given air excess involves little or no increase of temperature. In fact, in some of Riehm's experiments with a very large amount of supercharging there was an increase in power with a decreased percentage of air required for the combustion of the oil—that is, an increased air excess. This resulted in decreased exhaust temperatures and a decreased jacket-water loss.

If, however, advantage is taken of the fact that the use of supercharging also permits of a decrease in the air excess, increased temperatures will of course result. This gives stress and cooling problems which must be properly faced.

If a Diesel engine of a given design is barely able to deliver its maximum power with no trouble with valves, cylinder heads, or other parts, it would, of course, be futile to attempt supercharging of this kind. However, the fact that a number of engines are successfully operating with supercharging is prima facie evidence that Diesel engines can be designed to withstand the increase in the amount of power for a given cylinder displacement. There is no doubt that there are types of engines now in use which would give trouble if supercharging were attempted, but the success of other types with superchargers will lead to such alterations as will permit of a successful solution.

During the work done on the supercharging of airplane engines in the United States the supercharging has been pushed beyond the point of obtaining normal power at sea level and some supercharging has been done at low altitude, which has been called "supercharging below sea level." This, for an Otto-cycle engine, is the same thing proposed here for a Diesel engine, and the use of special steel alloys has permitted the successful operation of valves in these airplane engines at temperatures much greater than can ever be encountered in a Diesel engine.

In fact, it may be safely said that it is eminently practical to supercharge with a pressure of a few pounds per square inch, with an increase of clearance volume so as not to increase the compression pressure, and with but slight diminution in the air excess so as to avoid excessive exhaust temperatures. It is to be hoped, however, that future researches will permit of supercharging with pressure of 5 pounds per square inch or more, without increase of clearance volume but with increase of compression pressure and with increase of oil supply to the maximum amount which will give a clean exhaust. This will incorporate in one engine all of the advantages which have been gained independently, and will give a truly astonishing gain in thermal efficiency and power for a given engine weight.

Supercharging Two-Cycle Diesel Engines. Some manufacturers of Diesel engines have such valve arrangements as will give opportunity for entrance of scavenging air after the exhaust port is closed by the piston on the up stroke, and this has been called supercharging. However, this arrangement or any other arrangement heretofore used has little more than atmospheric pressure in the cylinder when the port was finally closed and compression begun. This is shown by the fact that two-cycle engines with the most effective scavenging give a mean effective pressure for the whole stroke of about 122 pounds per square inch. The lower end of the stroke is, of course, lost in a two-cycle engine, so that the effective stroke is about 77 per cent. This gives an equivalent mean effective pressure of 158 pounds per square inch during the effective stroke, which corresponds to values obtained by a good four-cycle Diesel engine. This means that the best two-cycle engine has a charge at the beginning of compression equal to that of a four-cycle engine. There must be a greater charge than this to give actual supercharging. This would require greater pressures and volumes than have ever been provided with reciprocating blowers, but these could easily be provided with centrifugal compressors. If the cylinder head, piston, and cooling arrangements can be constructed to stand the increased power, the centrifugal compressor can be arranged so that the air pressure in the cylinder at the beginning of the compression stroke will be above atmospheric pressure. This will result in the same increase of power due to supercharging which is discussed elsewhere for four-cycle engines. The higher pressure could be used for the scavenging also, which would then be done more expeditiously, so as to make a greater percentage of the stroke effective. It would also be possible to provide two centrifugal compressors, one such as is now used for scavenging, and a second one providing air at a higher pressure through a mechanical valve opened at the proper time.

EXHAUST-GAS TURBINES FOR SCAVENGING AND SUPERCHARGING.

In 1911 Dr. Wilhelm Schmidt proposed a centrifugal compressor driven by an exhaust-gas turbine for supercharging a four-cycle Diesel engine, and the proposal was repeated by Reihm. The same plan has since been extensively and successively used by Rateau in France and by the General Electric Company in the United States for airplane engines.

With this system the exhaust gases of a Diesel engine, instead of passing to the atmosphere, pass to an exhaust manifold and a nozzle box wherein is maintained a pressure above atmosphere, giving a back pressure against which the engine exhausts. The expansion of the gases through the nozzles from this back pressure to atmospheric pressure gives power which drives a turbine wheel. This turbine drives a centrifugal compressor which compresses air to a pressure above atmosphere and delivers it to the intake manifold. The engine displacement would then be filled with a charge at a pressure above atmosphere, just as in the case of a supercharger driven by a separate electric motor. In such a case the power for supercharging would be obtained by the decrease of the mean effective pressure shown by the indicator card, due to the increased exhaust back pressure. The theoretical efficiency is the same with a separately driven supercharger or an exhaust-gas-driven supercharger. The actual efficiency would depend upon the relative efficiency of transfer of power from the cylinder mean effective pressure, through the crankshaft, electric generator, and electric motor in the one case, and the efficiency of the turbine wheel in the other case. These two efficiencies are probably nearly equal. Data do not exist for a complete computation in the matter.

In the airplane installation the engine exhaust pressure is nearly equal to the supercharger pressure, and the difference in temperature enables the gas turbine to drive the compressor without mechanical connection to the engine.

It is to be expected that a similar situation would exist with a Diesel engine. The scavenging of the clearance space would be lost with this plan.

Exhaust-Gas Turbines with Extended Expansion. Modugno has put forward a number of other plans of using an exhaust-gas turbine with a Diesel engine. He proposes to obtain the "extended expansion" and theoretical gain due to increasing the length of the toe of the card, without supercharging, by gearing to the engine shaft an exhaust-gas turbine driven by increased engine-exhaust back pressure. He also proposes to obtain extended expansion as well as supercharging by using an exhaust-gas turbine with high back pressure which delivers considerably more power than necessary to drive a centrifugal supercharger.

The exhaust back pressure would then be greater than the supercharger pressure, so that these plans differ from Schmidt's exhaust-gas turbine by delivering power to the turbine wheel by raising the back pressure, which must be utilized by gearing. The author feels that in both of these plans the losses in the turbine wheel and gearing would be more than the theoretical gain due to the extended expansion.

Exhaust-Gas Turbine for Scavenging a Two-Cycle Diesel Engine. Figure 3 shows a plan devised by the author which has been made the subject of a few tests and an extended series of computations. An extra exhaust port is provided at the side of the cylinder somewhat in advance of the regular exhaust port. In this port is a check valve opening outward into a header which leads to the nozzle box of an exhaust-gas turbine. Four to six cylinders of a two-cycle engine must be connected to the header. When any one of the pistons passes the above-mentioned extra exhaust port during the expansion stroke, there will be sufficient pressure to open the check valve and pass a certain amount of exhaust gas to the nozzle box. This will continue until the piston begins to open the regular exhaust port. The pressure in the cylinder will then drop to a value below that in the nozzle box and the check valve for this particular cylinder will close. The exhaust will then pass out through the regular exhaust port and air will enter through the regular scavenging port just as in an ordinary two-cycle engine.

During the upward or compression stroke the pressure in the cylinder will be less than at the corresponding point in the down or expansion stroke, and the pressure in the nozzle box from other cylinders will prevent the check valve from opening. Hence, from the time that the check valve closes on the down stroke, the events in the cylinder will be exactly the same as for any two-cycle engine.

Thus, as each piston passes the extra exhaust port a charge of exhaust gas will pass into the nozzle box. With a six-cylinder engine there will be six such charges admitted during each revolution, and this will maintain a somewhat fluctuating pressure in the nozzle box. The magnitude of this pressure will, of course, depend upon the point in the stroke at which the extra exhaust port is placed. This point can be made anything which is found necessary in order to produce such pressure as will give the necessary amount of power for scavenging. As stated, the pressure in the nozzle box will fluctuate somewhat due to the successive charges admitted. This pressure is bound to be low enough during the time when each piston passes its extra exhaust port to permit the check valve to open, and the nozzle-box pressure will also be high enough to close the check valve as soon as the regular exhaust port is opened and to keep it closed during the rest of the down stroke and the early part of the compression or upstroke.

There will be provided a nozzle box, turbine wheel, and centrifugal compressor similar to those of an airplane supercharger. The proportions will be different, but the general design, materials, and methods of construction will be the same.

A certain amount of power is required to operate the scavenger for any two-cycle Diesel engine, and this power, of course, must be subtracted from the gross power which would otherwise be delivered from the engine. In the case of the present method this power is taken from the indicated horsepower. At the time the piston passes the extra exhaust port there will be an appreciable pressure in the cylinder, and the opening of this exhaust port will, of course, reduce this pressure below the value which would otherwise have existed during the remainder of the expansion stroke. An engine with an equipment of this kind will therefore have a jog in the expansion line near the end of the stroke which will make this line somewhat lower than it otherwise would have been down to the beginning of the exhaust. There will be an area between the expansion line as so lowered and the ordinary expansion line which will represent the power required for scavenging. This power will not pass through the piston rod and crankshaft as in the case of a direct-driven reciprocating scavenger, nor will it have to be produced by a separate generator and motor, with consequent losses as in the case of a separate centrifugal compressor. There will be instead losses due to some pressure drop through the check valve and the losses of the turbine wheel. Some computations made on this matter indicate that all of these different kinds of losses practically counterbalance each other, and that this method of scavenging will produce about the same amount of net power from Diesel engines as any other method. The proposed method is expected to give a great saving of weight and space, due to the fact that the apparatus consists merely of a turbine wheel and centrifugal compressor running at 10,000 to 20,000 R.P.M.—"Mechanical Engineering," Mid-November, 1925.

NEW DEVELOPMENTS IN HIGH VACUUM APPARATUS.

By G. L. Kothny.

The recent wide discussion and general interest in the use of high pressure and high temperature steam have brought into prominence the matter of closed feed-water systems. In these the condensate is preheated to within 20 to 50 degrees of the evaporation temperature by means of stage heaters.

Since the inter- and after-condenser of an air ejector are excellently fitted to serve as a stage heater, the use of steam air ejectors with inter- and after-condensers was given more consideration and brought about further development of this type.

By arranging an inter-condenser between the first and second stage the steam consumption of the two-stage ejector can be reduced about 50 per cent. The live steam issued from the first stage is condensed in the inter-condenser and the second stage has to remove only the saturated air vapor mixture from the inter-condenser.

The design of an inter-condenser requires the following considerations:

1. The ratio of air to steam is about 1000 times larger than that in the main condenser.
2. Due to this condition, different principles must be applied in its design.
3. The air vapor mixture withdrawn by the second stage must be absolutely free from entrained water globules.
4. This necessitates substantially complete separation of the condensate from the air undergoing compression.

A surface condenser of the ordinary type with water flowing through the tubes and the steam and condensate around them is not so well adapted to meet these conditions. With narrowly spaced tubes the danger exists that the condensate dripping over the outer surface of the tubes will be carried into the second state, unless a separator is arranged. Widely spaced tubes, however, mean low air velocity and consequent low heat transmission.

In Figure 1 a steam air ejector with surface inter-condenser of the Radojet type is shown. The design of the inter-condenser is original and embodies the considerations outlined above.

The air to be compressed enters into the suction chamber of the first stage, where it is acted upon in known manner by the steam issuing in jets from tubular nozzles. The mixture of air and steam is delivered through the

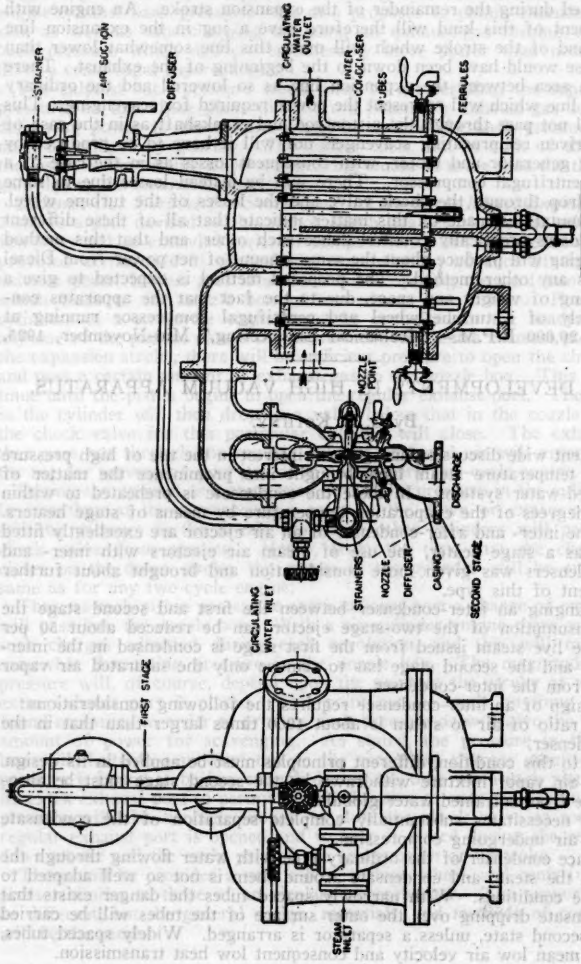


FIG. 1.

diffuser into the upper chamber of the condenser and passes downward through the first group of condenser tubes around which is flowing the cooling water, condensing most of the steam. The condensate remains in the lower chamber and passes off through a pipe into the drain control, while the air and steam continue their passage upward through the second group of condenser tubes, where the mixture is further cooled and a further portion of the vapor is condensed. The condensate drips downward through drain tubes into the lower chamber. The air and remaining vapors descend through a third group from the upper chamber into the lower chamber (the condensate passing off through drain pipes into the drain control), and from there pass upwards through a fourth group of tubes and the upper chamber into the suction chamber of the second stage, where they are compressed to above atmospheric pressure in the well-known manner.

Any condensate collecting on the upper tube sheet runs off through drain tubes which at the lower end have a water seal, thus maintaining the upper and lower chambers isolated from each other as to their pressure conditions.

The arrangement of a vertical surface condenser between stages of a multiple stage ejector with the air passing through the tubes have the following advantages:

1. Complete separation of the condensate from the air is obtained.

2. The velocity of the air passing through the tubes can be controlled by the arrangement of a certain number of tube passes, and thereby the best possible heat transfer can be obtained.

3. The steam jet issued from the first stage flows parallel to the tubes, instead of at right angles, as in ordinary surface condensers. This eliminates erosion of the condenser tubes, which is a common occurrence in surface inter-condensers of the ordinary type. If sufficient baffle plates are arranged to prevent erosion, such baffles cause a pressure drop in the inter-condenser, with a subsequent loss of efficiency in the performance of the ejector.

The principal factors controlling the performance of an air ejector with inter-condenser are:

(*a*) Condition of steam supplied.
(*b*) Back pressure at the discharge.
(*c*) The condition of the air handled.
(*d*) The inlet temperature and quantity of circulating water.
(*e*) Cooling surface of inter-condenser.

The effects of these factors on two-stage ejectors with inter-condensers will now be considered.

Regarding the steam supplied, it may be said that it is essential to use high pressure and dry saturated or superheated steam. The latter offers advantages for the inter-condenser type ejector because the steam from the first stage is condensed and the air to be compressed in the second stage is cooled in the inter-condenser. Higher steam pressure will increase the air handling capacity at vacua below 29 inches.

The effect of the back pressure is the same as on two-stage ejectors without inter-condensers. Back pressures of more than half-pound gauge should be avoided.

The effect of the condition of the air vapor mixture to be compressed has been the subject of further investigations. From scientific tests made at the Engineering Experiment Station at Annapolis (see A. S. N. E. JOURNAL, Vol. XXXII, No. 3), it has been found that although the general shape of the characteristics of an air ejector with inter-condenser handling saturated air vapor mixtures are the same as when handling expanded atmospheric

air under certain conditions, the weight of the saturated mixture handled is greater and, under other conditions, is less than the weight of expanded atmospheric air handled under identical operating conditions.

The inlet temperature of the circulating water as well as the quantity of the same will influence the performance of and the vacuum in the inter-condenser, which in turn again will influence the performance of the whole

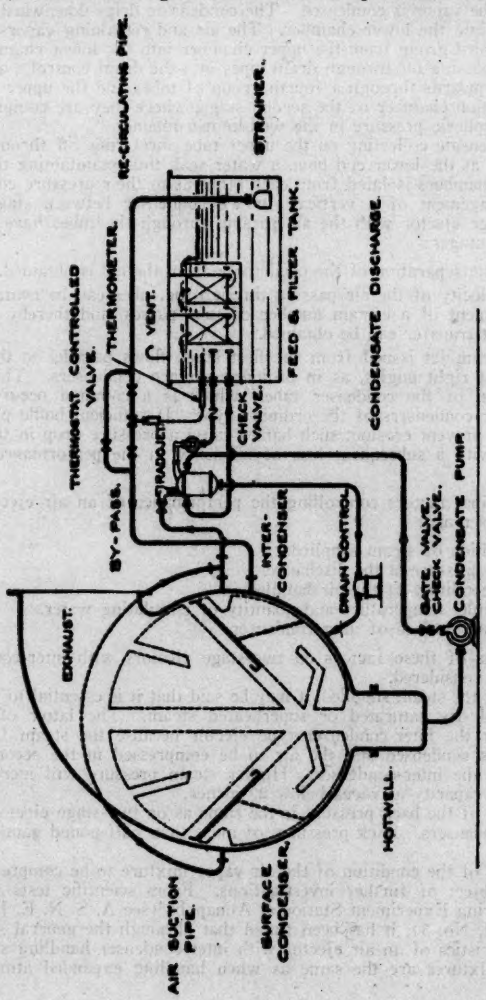


FIG. 2

ejector. The inlet temperature and the quantity of the circulating water must be within the practical limits of the desired vacuum. While it is possible to produce a vacuum of 29 inches with circulating water entering the inter-condenser at 100 degrees F., the amount of air handled will be less than that, if circulating water at 80 degrees F. or at lower temperature is supplied.

A typical arrangement of an air ejector with inter-condenser of the Radojet type for an open feed-water system is shown diagrammatically in Figure 2. The condensate from the main condenser is used as circulating water in the inter-condenser. During periods of insufficient supply of condensate when starting or maneuvering a thermostatically operated valve opens and re-circulates condensate from the feed tank into the main condenser, wherein it is cooled by falling over the tubes and returned through the inter-condenser into the feed tank.

In some installations the water side of the inter-condenser is divided into two sections, each of which has its own water supply. One receives condensate from the main condenser and the other cold fresh or salt water from another source. The cold water supply is thermostatically controlled by the temperature of the condensate leaving the inter-condenser.

A further development has been made by combining the inter- and after-condenser in one shell and thereby creating a compact and complete unit which entirely eliminates the absorption of air and non-condensable gases by the condensate when returning practically the total heat contained in the steam for extracting the air and vapors.

The arrangement of a steam air ejector with combined surface inter- and after-condenser in connection with a closed feed water system is diagrammatically shown in Figure 3. It will be noted that the condensate from the main condenser is returned directly into the boilers without coming in contact with the atmosphere. Absorption of oxygen is entirely avoided. The make-up feed is deaerated in the main condenser before entering the boilers.

Another development which should be recorded is that of the low-pressure steam-operated vacuum augmenter. The vacuum augmenter was invented by Sir Charles A. Parsons in 1903 and has since that time been applied in connection with wet air pumps of all types for the purpose of improving the vacuum created by the latter. High-pressure live steam was used as motive fluid in the augmenter, and hardly any changes in its design had been made since its invention until 1917, when, due to the incentive of Mr. John Metten, several augmenters of the Radojet type operated by exhaust steam of 3 to 5 pounds gauge pressure were developed and successfully installed and operated in connection with twin-beam air pumps on a number of torpedo-boat destroyers. It was found that, for the same ratio of compression and when handling the same amount of air, this new type of low-pressure Radojet augmenter required of exhaust steam only about 25 per cent of the quantity of live steam used by the Parsons augmenter.

Since then further improvements in the steam economy and a large number of applications on all types of steamers in connection with twin-beam air pumps have been made, among which are the passenger steamers *Yale, Harvard, Resolute, Reliance,* etc., U. S. scout cruisers *Trenton, Richmond, Marblehead, Memphis,* numerous torpedo-boat destroyers, freight steamers and ferryboats.

A low-pressure augmenter of the Radojet type, in connection with a twin-beam air pump, is diagrammatically shown in Figure 4. In this arrangement the heat from the exhaust steam, used for the operation of the augmenter, is returned to the boiler by circulating the condensate through the augmenter condenser. The latter is of special design and is provided with a separate cold pass through which circulating water is flowing when there is

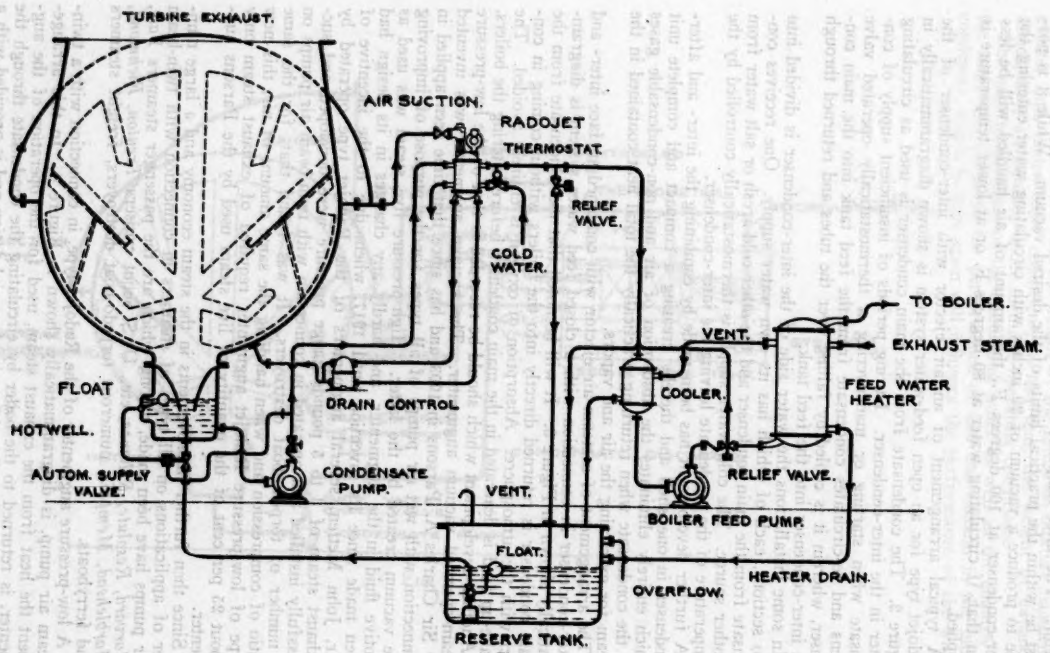

TURBINE EXHAUST.
AIR SUCTION.
RADOJET
THERMOSTAT.
RELIEF VALVE.
COLD WATER.
TO BOILER.
VENT.
EXHAUST STEAM.
FLOAT
DRAIN CONTROL
COOLER.
FEED WATER HEATER
HOTWELL.
AUTOM. SUPPLY VALVE.
CONDENSATE PUMP.
VENT.
RELIEF VALVE.
BOILER FEED PUMP.
FLOAT.
HEATER DRAIN.
OVERFLOW.
RESERVE TANK.

FIG. 3.

not enough condensate available from the main condenser. The cold water supply is regulated thermostatically or manually. The application of the low-pressure augmenter improves the vacuum from 1½ to 2 inches over that obtained by the twinplex or twin-beam air pump. An example of the saving in steam consumption resulting from the application of the low-pressure augmenter is given below:

Vacuum obtained by twinplex pump alone..... 3.75" hg. abs.
Vacuum obtained by twinplex pump and low-pressure augmenter in use..... 1.73" hg. abs.
Net saving in steam consumption of main turbine at full load. 9.1 per cent.

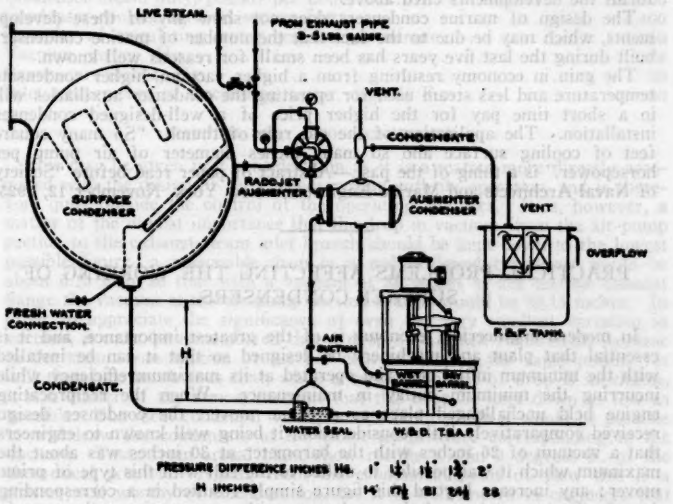


FIG. 4.

For the removal of the condensate from the high vacuum condenser turbine driven pumps of the centrifugal type are employed. If the total head, including the head due to the vacuum, exceeds 80 feet, two-stage type centrifugal pumps have to be used.

The turbine driving the pumps is equipped with a constant speed governor, an emergency trip governor and a hand-operated valve for admitting steam to an additional nozzle in case the boiler pressure falls considerably below normal. In low vacuum condenser installations the horizontal duplex reciprocating type piston pump is used.

There has been little recent improvement in circulating pumps for condenser service, due to the fact that pumps of the centrifugal type have reached a stage of development which is very near to the practical limits of high efficiency.

The development in the design of surface condensers has been confined to condensers used in stationary power plants and has been along the lines

of increasing heat transfer and vacuum, deaeration of the condensate, highest possible condensate temperature, equal steam distribution and minimum pressure drop between exhaust opening and air pump suction.

In stationary power plants an absolute pressure of 1.3 inches with 70 degrees circulating water inlet temperature and a condensate temperature of 87 degrees F. corresponding to the absolute pressure at the exhaust inlet are being obtained. This is accomplished by proper spacing of the tubes and providing lanes leading exhaust steam to the bottom of the condenser for heating the condensate and assisting in its deaeration. Equal steam distribution is obtained by providing several air pump suctions, which also reduce the pressure drop and prevent inactive tube surface. There are a number of different tube sheet layouts in use which all embody in various forms the developments cited above.

The design of marine condensers does not show any of these developments, which may be due to the fact that the number of marine condensers built during the last five years has been small, for reasons well known.

The gain in economy resulting from a higher vacuum, higher condensate temperature and less steam used for operating the condenser auxiliaries will in a short time pay for the higher price of a well-designed condenser installation. The application of the old rule of thumb: "So many square feet of cooling surface and so many inches diameter of air pump per horsepower," is a thing of the past.—Abstract of paper read before "Society of Naval Architects and Marine Engineers," New York, November 12, 1925.

PRACTICAL PROBLEMS AFFECTING THE WORKING OF SURFACE CONDENSERS.

In modern engineering, economy is of the greatest importance, and it is essential that plant and machinery be designed so that it can be installed with the minimum initial cost and operated at its maximum efficiency while incurring the minimum outlay in maintenance. When the reciprocating engine held unchallenged place as a prime mover, the condenser design received comparatively little consideration, it being well known to engineers that a vacuum of 26 inches with the barometer at 30 inches was about the maximum which it was possible to utilize to the full with this type of prime mover; any increase beyond this figure simply resulted in a corresponding drop in the temperature of the condensate with a proportionate increase in the coal consumption, without any compensation being secured in the form of increased power from the engine. The inability to utilize high vacua being directly associated with the fundamental principle and design of the reciprocating engine, condenser designers were not called upon to meet exacting conditions. With the advent of the steam turbine as a commercial proposition, however, this state of affairs was changed, and an entirely different proposition was presented, the fundamental principle on which this prime mover is based being such that it can be designed to make full use of the highest vacuum obtainable. Assuming an air pump of ample capacity has been installed, the determining factor will be the temperature of the cooling water to the condenser. In marine work this will, of course, be the sea temperature, and that in turn will, obviously, vary with the geographical position of the ship and the corresponding climatic conditions obtaining. In the interests of economy it is therefore a matter of supreme importance that the vacuum maintained in the condenser of a steam turbine plant should be the absolute maximum which it is possible for the plant installed to maintain under the prevailing conditions.

The three most important factors in the economical maintenance of high vacua are:—

(1) That the vacuum drop across the condenser should be kept down to an absolute minimum.
(2) That the temperature drop across the condenser should be kept down to an absolute minimum.
(3) That the minimum power be absorbed in operating the condenser auxiliaries.

As a basis for considerations in the course of this article, the following conditions will be assumed:—

Power developed, S.H.P.	3,000
Condenser steam duty, pounds per hour	37,000
Condenser cooling surface, square feet	4,400
Quantity of circulating water, gallons per minute	5,500
Vacuum, inches, Hg.	28
Barometer, inches, Hg.	30
Inlet temperature of circulating water, degrees F.	80

VACUUM DROP ACROSS CONDENSER.

This may be given first consideration as it is a matter more directly associated with the design of the condenser than with its operation, and is therefore quite outside the control of the operating engineer. It is, however, a matter of the utmost importance that the drop in vacuum from the air-pump suction to the exhaust-steam inlet branch should be kept down to the lowest possible figure; a reasonable drop in a well-designed condenser would be about 0.15 inch, so that with a vacuum of 28 inches at the turbine exhaust flange the vacuum at the air-pump suction branch would be 28.15 inches. In order to appreciate the significance of even the very smallest variation in vacuum, it must be remembered that a drop of 0.1 inch requires an increase in the steam consumption of 0.6 per cent to develop the same power. It will therefore be obvious that, while a certain amount of drop is inevitable, it is a matter of the utmost moment that this drop be kept down as low as possible. The ideal condenser would be one from which it would be possible to withdraw the condensate at the exact temperature corresponding to the vacuum. Tests from actual plant show results within 3 degrees F. of the theoretical possible. Again, from the economical point of view this is a matter well worthy of consideration, for it is well known to engineers that a drop of 10 degrees in the temperature of the condensate approximates closely to an increase of 1 per cent in the coal bill. In actual marine practice the drop frequently amounts to 7 or 8 degrees, and this, it will be seen, is equivalent to an increase in the coal bill of from 0.3 to 0.4 per cent beyond that unavoidably lost by the 3-degree temperature drop mentioned above. The loss thus incurred may frequently be attributed to passing too much circulating water through the condenser for the particular quantity of steam being dealt with at the time, condensation having been completed in the upper portion of the condenser, while the condensed steam, falling over the bottom tubes through which the coldest water is passing, is still further reduced in temperature, and B.T.U.'s are thus thrown to waste in the outgoing circulating water. If the temperature difference is allowed to increase from 3 degrees to, say, 7 degrees F., the extra loss of heat in a plant having a steam duty similar to the set at present under consideration will amount to 148,000 B.T.U. per hour, which corresponds, roughly, under average conditions, to £70 per annum. The vacuum efficiency of a condenser is the ratio:—

$$\frac{\text{Actual vacuum obtained at turbine exhaust flange}}{\text{Vacuum corresponding to the condensate temperature}}$$

from which it will be seen that the ideal condenser from which the condensate would be withdrawn at a temperature exactly equal to the steam would, of course, have a vacuum efficiency of 100 per cent. In the present case, with a vacuum of 28 inches and assuming the condensate to leave the condenser at a temperature 3 degrees below that of the steam, the vacuum efficiency would be

$$\frac{\text{Actual vacuum} = 28 \text{ in.}}{\text{Temp. } 98.15 \text{ deg.} = 28.18 \text{ in.}} = 99.4 \text{ per cent.}$$

MINIMUM POWER TO AUXILIARIES.

It has already been shown that if too much circulating water be pumped through the condenser a very considerable heat loss is incurred; to this loss must be added the extra power absorbed by the pump in delivering the extra water through the condenser, and this being quite an appreciable amount, is a matter worthy of consideration. In view of the double loss incurred in this way, it is a matter of the utmost importance that only the requisite amount of water be passed through the condenser, and this can be checked in two ways, the first being the condensate temperature drop, causing a drop in the vacuum efficiency, as previously explained, while the second is the comparatively low temperature of the outgoing circulating water.

$$\text{The ratio} \frac{\text{Actual vacuum obtained at turbine exhaust flange}}{\text{Vacuum corresponding to outlet circulating temp.}}$$

is the condenser efficiency. At this stage it might be worth while to draw attention to the difference between the "vacuum efficiency" and the "condenser efficiency"; the former shows the relationship of the actual vacuum obtained at the turbine exhaust flange to the condensate temperature, while the latter is based on the outgoing circulating water.

When fixing up the size of the condenser for a particular steam duty, two courses are open to the designer—one is to put forward a plant which will meet the vacuum and temperature conditions specified in the most economical manner from the engineering point of view; the alternative scheme would have more regard for initial cost, weight, and space taken up by the plant. In either case the vacuum and temperature drop can be kept down to a very low figure in a properly designed condenser, but in the latter the power absorbed by the circulating pump goes up as the quantity of water passed through the condenser has been increased in order to reduce the surface and thus secure a smaller, lighter and less costly condenser. In marine work, where an abundance of cooling water is available for circulating purposes, and where it is distinctly advantageous to have machinery as small and light as possible, the condenser is usually designed to pass a relatively large quantity of water and to have a comparatively small cooling surface. This means that the plant will have a low condenser efficiency. A condenser having an efficiency of 100 per cent would be one in which the vacuum maintained would be equal to the theoretical vacuum corresponding to the temperature of the outgoing circulating water; in other words, the outlet temperature of the circulating water would be equal to the steam temperature, as in that case the quantity of water passing through the condenser would be the absolute minimum capable of absorbing the heat given up by the steam. In practice a condenser having an efficiency of 100 per cent would present a very unsatisfactory proposition, for it will be apparent that when the circulating water was about to leave the condenser its temperature, being equal to that of the steam on the opposite side of the tube, would make it impossible for transference of heat to take place, and, as a consequence, the upper portion of the condenser would cease to function. In order to obtain the necessary difference in temperature between the steam and the water,

the latter may be arranged to pass out at a temperature of not less than 6 degrees F. below that of the steam. In marine work, where plenty of cooling water is available, and also where it is distinctly advantageous to reduce the weight to be carried as much as possible, the difference in temperature is usually very much larger, the outlet temperature being as a rule about midway between the inlet temperature and the steam temperature. In the plant at present under consideration, the vacuum being 28 inches, the steam temperature will be about 101 degrees F., and with the circulating water at an inlet temperature of 80 degrees, the outlet temperature may be taken at about 91 degrees F. The vacuum, corresponding to a temperature of 91 degrees, is just a little over 28.5 inches; therefore, the condenser efficiency in this case will be about 98 per cent. A falling-off in the vacuum accompanied by a drop in the condenser efficiency serves to indicate that the condenser tubes have become dirty, resulting in a lower rate of heat transmission being maintained.

STRAINERS.

It is frequently the case that in rivers trouble is experienced due to extraneous matter of a large variety entering with the circulating water. In naval work and in some of the larger steamship companies it is the practice to install strainers in the form of perforated plates, usually in duplicate, so that cleaning can be effected without interrupting the main supply, these strainers being, of course, in addition to the usual grids on the ship's side. When the water is particularly bad in this respect, the strainers prove of very great assistance in preventing the choking up of the condenser, and consequently reduce the frequency with which it would otherwise be necessary to remove the inspection doors on the end covers at the inlet end, in order to clean the tubeplate which would have become the strainer, to all intents and purposes. In land work, straining the circulating water presents a very real problem, and as a result of the attention which this subject has demanded, there are on the market a large number of types and designs of straining devices, from the ordinary stationary cage type to the more elaborate machanically operated disc with automatic flushing and cleaning arrangements. When the effect of even a small drop in the vacuum is considered, it will be appreciated that the initial cost of installing straining apparatus, and the expense involved in its operation and maintenance where power is absorbed, is more than justified if it results in keeping the condenser in a state of maximum efficiency and maintaining the highest possible vacuum. In tramp steamers and vessels having reciprocating engines generally, this difficulty does not crop up in the same acute form, a vacuum of 26 inches Hg. being secured without any great difficulty, assuming that the conditions, temperature of cooling water available, air leakage into the exhaust system, size of condenser and capacity of air pump, are reasonable.

CLEANING THE CONDENSER.

The rate at which the condenser tubes become fouled will depend on the composition of the water being used for circulating purposes, vessels sailing in tropical waters and those frequently on river work being particularly subject to trouble in keeping their condenser tubes clean. The problem of maintaining the vacuum becomes more acute as the deposit of slime and mud or hard scale on the inside of the tubes becomes thicker, steadily increasing the resistance to the passage of the heat from the exhaust steam to the circulating water. One theory advanced as a means of preventing the deposit of foreign matter on the inside of the tubes is that of passing the water through at a relatively high velocity; it is claimed that by this means the washing or scrubbing action of the water keeps the inner surface clean.

That this method is to some extent successful in achieving the aim in view will be generally conceded, but whether the results are entirely satisfactory, especially when considered from the point of view of the longevity of the tubes, is at present a matter of considerable controversy. In the event of the tubes becoming partially clogged with deposit, the cleaning process may be effected either by the more mechanical methods of scraping and brushing or by the chemical process of circulating through the tubes suitable cleaning agents which attack and dissolve the extraneous matter, after which the solution is withdrawn and the condenser washed out. The former method is probably less expensive so far as material cost is concerned, but it involves the holding up of the plant for cleaning operations for a very much longer time than that required by the chemical process. The nature and strength of the cleaning liquid and the length of time it is necessary to continue the process depends on the form and quantity of deposit to be removed from the tubes. When it is of the muddy or greasy variety, a solution, of which the constituents are 1 pound of caustic soda to 100 pounds of water, can be used with advantage. If it is found necessary, the percentage of caustic soda may be increased up to 7 to give a stronger cleaning solution. When the extraneous matter takes the form of hard scale, a most efficient cleaning agent is a dilute solution of hydrochloric acid. The strength of the mixture required will again depend on the amount of deposit to be removed. In cases where the condenser is not in a particularly bad state, the cleaning may be effected with a solution of 5 parts hydrochloric acid to 100 parts water, but when the conditions call for a more powerful cleanser the percentage of hydrochloric acid may be increased up to, but not beyond, 40 per cent. With either solution, hydrochloric acid or caustic soda, the temperature must not be allowed to rise too high, 150 degrees F. being the maximum, and in no case should this be exceeded. The process of cleaning the condenser in this way consists of closing the circulating water valves and almost filling the condenser with water; some estimate of the quantity of water will require to be made, as this information will be necessary in determining the required amount of cleaning liquid to be added to obtain a solution of the correct strength. By means of a steam jet the water is caused to circulate through the tubes, and when this state is established the cleaning liquid is introduced.

Some of the engineering firms who specialize in condensing plant and the problems associated therewith manufacture apparatus specially designed for this process. The cleaning equipment, which may be arranged to fit on to the end covers of the condenser, consists of two connections into the water space, one vertically above the other, each fitted with a cock and connected up by a pipe; at the lower end a steam jet is arranged so that when the cocks are opened and steam turned on the jet will issue through the lower connection into the water space and thereby set up a circulation in a direction away from the lower cock into the condenser, and through the tubes, while water will be drawn over through the top cock, down the connecting pipe, and pass back into the condenser by the action of the steam jet; in this way circulation is set up and maintained. Should there be any tendency for the cleaning solution to become too hot due to the condensation of the steam, the jet must be shut off for the length of time necessary to insure that the temperature shall not exceed 150 degrees F. At the upper end of the connecting pipe a filler with a small cock is fitted to receive and control the admission of the cleaning liquid. It may be found necessary to continue the operation for several hours, but if the cleaning process is carried out with regular frequency a much shorter time will suffice. After the operation has been completed the solution should be drained off and the condenser thoroughly washed, in the case of caustic soda, with water; when hydrochloric acid has been used the condenser must be washed out several times with an alkaline solution. The advantages which may be claimed for this

method, applied by means of efficient apparatus, are: (1) That by keeping the cooling surface clean, the maximum rate of heat transmission is maintained and the performance of the condenser continues to be that of a highly efficient unit. (2) The time required to clean the condenser in this way is only a very small fraction of that taken up in removing end covers and cleaning the tubes by mechanical means. This is an outstanding feature of this particular method, the first advantage being that secured by maintaining a clean condenser quite irrespective of the method employed to keep the cooling surface in that desired state. (3) For well-designed apparatus it can be claimed that it is small and compact, taking up very little space, and, being permanently secured to the end covers, cleaning operations can commence without any elaborate preliminary preparations.

CONDENSER TUBE CORROSION.

This subject is a vast and most complicated one, on which the information at present available is comparatively scanty, although valuable research work is constantly being carried out in the matter. The contributory causes to a state of active corrosion in the tubes of a surface condenser are numerous, involving such diverse subjects as the chemical constituents of the circulating water, the composition of the tubes, the velocity of the water through the tubes, and the mechanical design of the water heads and ferrules. The detrimental effects are also numerous, entailing, for example, the cost of supplying new tubes more or less frequently, or perhaps, in very bad cases, completely retubing the condenser in an unwarrantably short period of time, the ship or plant being held up while this operation is being carried out; another effect is that of leaky tubes, causing contaminated condensate with the manifold troubles that inevitably accompany impure feed water. The method of attack against corrosion may be divided into three main sections, the first being to treat the water itself in such a way as to render it noncorrosive; the second is associated with the composition of the tubes and includes numerous attempts to discover a suitable alloy which would be capable of resisting the action of untreated water. As an auxiliary to this method, attempts have been made to discover a suitable protecting solution with which the ordinary tubes could be coated and thus rendered proof against the ill-effects of raw water; in the third method, the attempt is neither to deal directly with the tubes nor with the water, but by means of an electrical current to protect the tubes against the action of the water by installing slabs of metal in such a way that the corroding action will take place on them.—"Marine Engineer and Motorship Builder," November, 1925.

NAVY INVESTIGATES ULTRA FREQUENCIES.

By Dr. A. Hoyt Taylor.

THE RESULTS OF EXPERIMENTS CARRIED ON BY DR. TAYLOR UNDER THE AUSPICES OF THE NAVY DEPARTMENT, AT THE ANACOSTIA, D. C., EXPERIMENTAL STATION, AND THE RECENT EXPERIMENTS OF THE N. R. R. L. ARE TOLD IN THE FOLLOWING ARTICLE.

The object of this article is to present graphically and in systematic form information which is a summary of the range data for various frequencies, so far as it can be estimated from the extensive experiments carried on by this laboratory, supplemented by considerable information from outside sources which has come in various ways to our knowledge.

Two further objects of the article are, first, to indicate the regions which require further exploration and, second, to show the places where certain transition phenomena of a more or less abrupt character occur as the fre-

quencies are varied from 100 kilocycles to 20,000 kilocycles. This should bring out the peculiarities of high frequency transmission and serve as a guide in a general way in formulating policies looking forward to the possible wider adoption of high frequency communication in the Naval service.

The range chart is based on the following considerations:

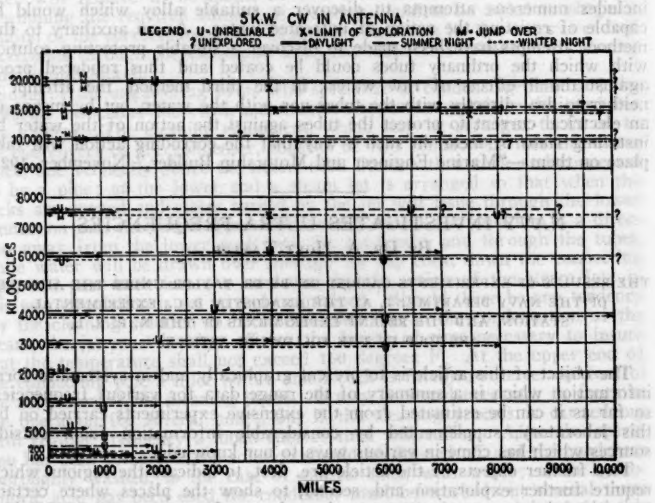
(*a*) Five kilowatts in the antenna.
(*b*) Average antenna installation.
(*c*) Communication between points on the same meridian.

The chart is, nevertheless, generally applicable to east and west communication or any communication where there is considerable time difference between the points involved, provided due accounts are taken of this time difference. Nevertheless, it must be admitted at the start that the problem is much more complicated for such a condition, especially where there is a very large number of hours of time difference between two points.

WIDE VARIATION IN NIGHT RANGE.

There is, of course, considerable difference between the daylight ranges, summer and winter, but not anything like the differences which occur in the night range. Therefore, the line on the chart indicating daylight ranges must be considered as average ranges, summer and winter, but for the night range, the lower dotted line indicates the winter night range and the upper dotted line indicates summer night range.

A cross entered on the line indicates the limit of actual exploration and the extension of the line beyond the cross indicates the probable range. A "V" entered on the line, indicates, if bound on right and left by two arrows, a region (generally a short region) within which communication is uncertain—or, in case the "U" stands farther out to the right-hand side of the diagram, it indicates that for ranges longer than those corresponding to the position of the "U" communication begins to become uncertain.



EFFECTIVE RANGE OF TRANSMISSION FROM 15 METERS TO 3000 METERS.

As an illustration, take the 4000-kilocycle band. The daylight communication is set at about 750 miles. The summer night communication is at about 7000, but uncertain after 3000. The winter night communication extends to 10,000 miles but is subject to some uncertainty after 6000. Another instance of the use of the "U" is shown in the 6000 kilocycle band in which there is an uncertain region between 150 and 400 miles beyond which the range again becomes certain and extends to 10,000 miles (probably) but is uncertain after 7000. That is, this frequency has two regions of uncertainty; one of close regions and the other at very distant regions. The use of the question mark ("?") is for the purpose of indicating unexplored regions. The use of the "M" on the diagram means that the radiation "skips over" or "misses" entirely the region indicated; therefore the "M" is always bounded by arrows right and left. An instance of this is the 15,000 kilocycle band which skips over the region from 75 miles to 700.

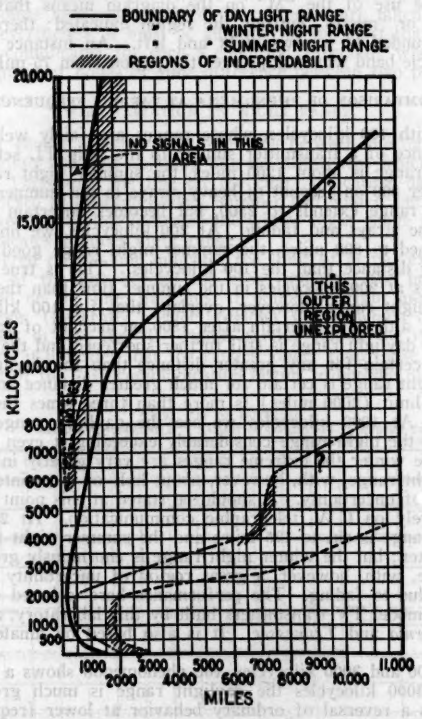
COMPARISON OF PHENOMENA AT VARIOUS FREQUENCIES.

Starting with 100 kilocycles, whose ranges are fairly well known from the performance of a transmitter similar to the Ship TL sets, we see that the daylight range is about 1200 miles, the summer night range 2000, but uncertain after 800 on account of heavy strays in the summer time, and the winter night range extends to 2500, but becomes uncertain after 2000 on account of the strays and fading. At 200 kilocycles, we find the daylight range shortened to 800 miles, the summer night range good on the whole for a greater distance than the 100 kilocycles. This is true because there are less strays at 200 kilocycles in the summer time than there are at 100. The winter night range, however, overlaps that for 100 kilocycles, going to 3000 miles, although uncertain after 1800 on account of fading. At 500 kilocycles the daylight range is still further shortened and the summer night range is not certain for any greater distance than the daylight range, but the winter night range is certain for much greater distance and the extreme winter night limit (2000 miles) is more than three times the normal daylight range. At 1000 kilocycles we see the daylight range still further shortened but the night range considerably exceeding it even in the winter, whereas in the winter the extreme ranges are very greatly in excess of the normal daylight range, with, however, about half of the winter night range in the region of uncertainty. It should be stated at this point that the table is based entirely on C.W. telegraphic communication. At 2000 kilocycles the daylight range is cut to 125 miles and the summer night range is not a great deal better; but the winter night range is enormously greater than the daylight range, with, however, a great region of uncertainty in the winter night range, due to fading. The performance here is based on the Fleet's report of the model TV transmitters built by this laboratory, and tested out on the *California* and *Tennessee*. It is also based on amateur data to a certain extent.

Between 2000 and 3000 kilocycles, the phenomenon shows a rather abrupt change. At 3000 kilocycles the daylight range is much greater than at 2000, which is a reversal of ordinary behavior at lower frequencies. The summer night range is enormously extended and the winter night range still more so. We see that the reliable night ranges for summer jump to 2000 miles and as a matter of fact this figure is probably considerably underestimated. It is, however, desired to make the chart conservative, at least in its application to the higher frequencies. The night ranges in the winter time, however, are certain up to 6000 miles. Comparing 3000 kilocycles with 100 kilocycles, we find the 100 kilocycles excels in daylight range but that the 3000 kilocycles greatly excels in possible night ranges. At 4000 kilocycles we see the daylight range extended to 750 miles, the summer night range certain to 3000, with a possible night range to 7000; while the

winter night range is certain to 6000 with possible ranges to 10,000. At 6000 kilocycles the daylight range extends to 1000 miles and the summer night range to 4000 with possible ranges to 7000, whereas the winter night range probably goes to 10,000 but has not been fully explored. We may consider it uncertain at least after 7000.

A new and interesting phenomenon makes its presence felt for the first time in the diagram regarding an uncertain period within short distances during the winter night transmission; namely, between 150 and 400 miles.



In the next line on the diagram for 7500 kilocycles, the daylight range has been further extended to 1200 miles, thus nearly equalling the 100 kilocycle transmitter but an uncertain region not far from the transmitter has been introduced between 100 and 350 miles during the summer night range and a skip, or entirely missed region, occurs in winter night ranges between 100 and 350 miles. This frequency has been explored to 4000 miles, but it is believed that it will carry very much further. We may say that it is uncertain, however, after 8000 and probably will carry to 10,000 on the winter nights. At 10,000 kilocycles we see that the "jump" or "miss" occurs both

summer and winter and also by daylight but the daylight jump is only about 500 miles, whereas the summer night jump is very great indeed. Very little exploration has been made of this frequency for nocturnal transmission but there is good reason to suspect that the summer night jump is in the neighborhood of 2000 miles and the winter night jump possibly 4000 miles. There is also good reason to believe that frequencies not far from 10,000 kilocycles can be successfully used for extreme night ranges. (Note Samoa's successful reception and steady intensity of tests from Schenectady, N. Y., on tests of 35 meters at night.)

Fifteen thousand kilocycles: Here the daylight jump is increased to between 600 and 700 miles and an uncertain region follows this to 1000 miles, but beyond this, so far as the exploration has gone (4000 miles), results are excellent. It is impossible to say what the daylight range will come to beyond 4000 miles. It is known, however, that the missing region or the jump in this frequency is very great at night, both summer and winter. Very little exploration has been made here, but there is some data indicating that this frequency can be successfully used at 10,000 miles, even at night. This statement is based on the establishment of two-way communication with Australian 2-CM (Sydney). He used 21 meters and this station used 20.8 meters, between 1 A. M. and 2:30 A. M. E.S.T. It was broad daylight when the test commenced. It should be noted at this point that this laboratory has, so far, not used more than 750 watts in the antenna in the twenty meter band. Australian 2-CM was using still lower power. It is evident then that figures estimated for 5 kilowatts in the antenna but actually based on experiments with less than 1 kilowatt, ought to be fairly conservative.

At 20,000 kilocycles the exploration is very scanty indeed, but there is information at hand giving the distances of the daylight jump as in the neighborhood of 1500 miles with an uncertain region extending to 2500 miles and which probably is followed by a certain region for a considerable distance beyond. Absolutely nothing, however, is known of night ranges on these frequencies. The ranges of the direct, or earth-bound components are well enough known; they are about 60 to 70 miles for 15,000 kilocycles and in the neighborhood of 40 to 50 miles for 20,000 kilocycles.

It is not the purpose of this particular report to go into details of the vast amount of information upon which the range chart is based, nor to divulge at this particular time the theory which is gradually forming in the minds of the engineers of this laboratory which we believe will account for these curious effects. The purpose of this report is to serve as a practical guide to indicate what ranges may be covered at different frequencies and what ranges remain to be explored, and what we hope to get in the unexplored regions.

It may be stated at the present time, however, that some of the most valuable information confirming earlier data on the matter of the "skip" or "miss" region was obtained from the daily reports made by Major J. O. Mauborgne, U. S. A., from the Army transport *St. Mihiel*, who took observations on 16, 32, 20.8 and 41.7 meters, all the way from New York to Panama. It is believed that the information concerning the uncertain regions and "skip" regions is fairly definitely known for daylight work. If any special criticism could be made of the range chart it would be that it underestimates the summer night ranges on 3000 kilocycles.

When one considers the chart as a whole, the high frequencies show clearly their enormous superiority from a point of view of economy on power consumption and general cost, and further it is possible to obtain ranges with high frequencies which we cannot hope to equal with almost any practical amount of power on lower frequencies.

EAST AND WEST COMMUNICATION.

To apply the chart to east and west communication, we must, for the present, consider that during the hours when daylight obtains over the entire stretch, we apply the daylight range data. For the hours which night obtains over the entire stretch, we apply the night data. In the intermediate hours when part is sunlit and part dark, much further exploration will have to be made, but we do know that a sort of compromise condition does exist and it does appear further that a 5-kilowatt transmitter, equipped with about four frequencies would be in a position to obtain highly creditable ranges at any time of either day or night, and whether for north and south communication, or for east and west. We must, however, at present, until further information comes in from stations like Samoa, Guam and Cavita, be forced to believe that the east and west problem is more difficult of practical solution.

It must be understood in referring to the range chart, that estimates on range and reference to the "missing" or "skipped" areas on the higher frequencies refer in the case of the daylight ranges to conditions existing in the middle of the day and for the night ranges, to conditions existing in the middle of the night. For west and east work this must be interpreted as meaning conditions when the sun is half-way between the two meridians under consideration. It is well known, of course, that there is a more or less gradual transition from daylight to dark conditions; in fact, it is not nearly as abrupt as one would anticipate it to be, especially in the summer time.

Since the first part of this report was written Samoa has reported successful reception of our 20.8 meter wave as early as 8 P. M. zone plus five time, which means that 6000 out of the 7000 miles between Washington and Samoa were traversed in daylight. Also, it is well known that 20 meter signals from amateurs on the west coast are now received as late as midnight or 1 A. M. zone plus five time, which could not have been done during winter nights. This is interesting as showing a gradual change in the skipped region for 20 meters. The skipped region is less in the daytime, gradually increases to somewhat less than 2000 miles in the summer nights and very likely is considerably in excess of this in the winter nights, although it is not known with certainty whether it ever comes down to earth again in the winter nights. One must therefore conceive of the skipped distance on the higher frequencies undergoing a lengthening process as the night wears on, followed by shortening process as daylight approaches. Most of the information on the higher frequencies must, of course, be considered as incomplete and subject to future revision. Nevertheless certain fundamental things in the behavior of these frequencies seem to be quite definitely established.

Balboa reports satisfactory reception of our 20.8 meters throughout the 24 hours during the summer but it is not anticipated that this will be possible in the winter time. Very likely the signals as received at Balboa will fade out during 6 or 7 hours during the winter nights. At the present time the 20.8 meters with less than 1 kilowatt in the antenna is more satisfactory for handling traffic with Balboa than Annapolis on 17,000 meters.

SUMMARY.

A preliminary range chart has been constructed for telegraphic communication, 5 kilowatts in the antenna, at various frequencies. The conclusions upon which the range chart is based, are derived from experiments made by the Naval Research Laboratory, from experiments made by amateurs and upon such data as the Laboratory has had access to from commercial and Government sources at home and abroad.

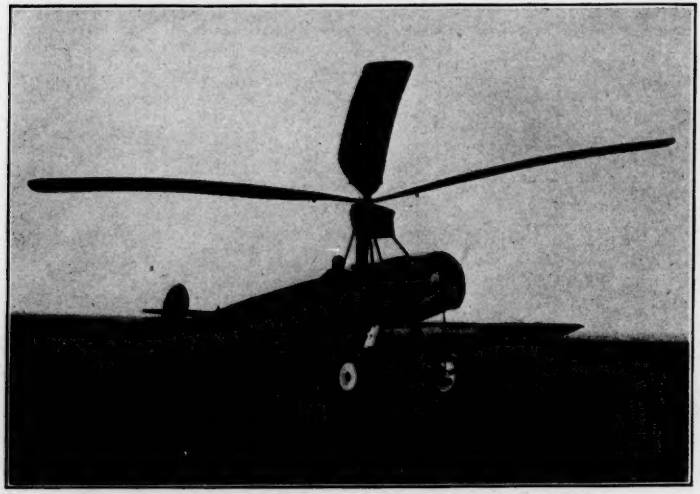


FIGURE 1.

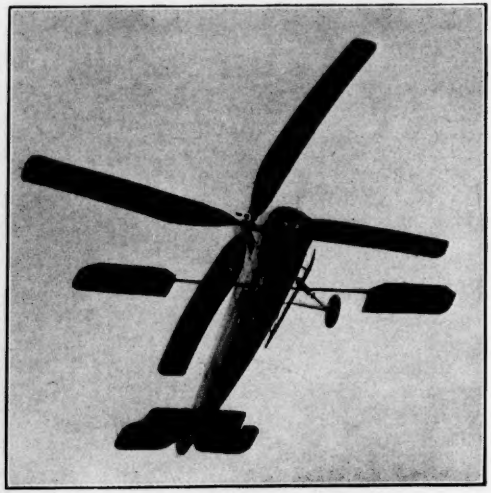


FIGURE 2.

An attempt has been made to indicate in a general way the advantages and disadvantages of high frequency telegraphic transmission. Various critical regions are pointed out where new phenomena make their appearance: in particular these regions are (1) the region between 2000 and 3000 kilocycles where daylight ranges begin to increase with increasing frequency at the same time that the night ranges show extremely great increase and a degree of reliability which would be wholly unanticipated from observations made at frequencies lower than 2000 kilocycles. (2) A region around 6000 kilocycles where an uncertainty develops during the winter nights at relatively very short ranges. (3) The development at successfully higher frequencies of this uncertain range into the missing region which is most pronounced in the winter nights. and finally as the frequency is increased makes itself felt in the summer nights and at still higher frequencies even in the daytime.

The development of this missing region to extensive areas is shown to take place with frequency rise to 20,000 kilocycles. The chart also attempts to indicate in a general way, the region of uncertain communication and the regions where further exploration is urgently needed. It is quite evident that the range data is far from complete and that many individual cases will be found in contradiction to the chart, but it does represent a sort of general average of the situation as it presents itself to the engineers in the Naval service.—"Radio News for January," 1926.

THE CIERVA FLYING MACHINE.

It is perhaps rather remarkable that the whole of the progress made in flying with machines of the heavier-than-air-type has resulted from a development of the first practical flying machine, produced by the Wright brothers in 1903, a modern aeroplane differing but little in its essentials from the earliest machine to fly successfully. Many attempts have, of course, been made to produce a practical flying machine of the helicopter type employing airscrews rotating in the horizontal plane to give a direct lift, and although such machines have risen from the ground, we believe that even their inventors would hardly claim that the problem of the helicopter has been solved. Some of the desirable characteristics of the helicopter have, however, been obtained in a machine which, although it depends for its support upon an airscrew rotating in the horizontal plane, cannot be classed as an orthodox helicopter, since it is unable to rise vertically from the ground and hover stationary in the air. It can, however, in addition to flying, climbing and gliding like an ordinary aeroplane, fly at very low speeds and descend almost vertically under complete control. The low-speed characteristic would appear to be useful for observation and other military purposes, while the ability to descend steeply would enable the machine to land without difficulty in a comparatively confined space, such as might be available in the center of a large city or on the deck of an aircraft carrier at sea.

The machine to which we are referring is the invention of Senor Don Juan de la Cierva, who brought it to this country on the invitation of the Air Ministry. A number of demonstration flights were carried out with complete success over Laffan's Plain, Farnborough, on Monday last in the presence of the Secretary of State for Air and other officials, the machine being piloted on this occasion by the well-known test pilot Capt. F. T. Courtney. Illustrations of the machine on the ground and in the air, reproduced from photographs taken by the Topical Press Agency during the course of the demonstration, are given in Figures 1 and 2 opposite and from these its general appearance can be seen. It consists, as shown, of an ordinary Avro fuselage with a 110 horsepower Le Rhone engine,

driving a tractor airscrew in the normal position. The rudder, tail plane and elevator are of the usual Avro design, but the wings are replaced by a large four-bladed airscrew, mounted on ball bearings on an almost vertical shaft, so that it is free to rotate in a horizontal plane. The blades of this airscrew are of symmetrical section (Göttingen 429) and are in external appearance similar to that of an ordinary aeroplane wing of exceptionally high aspect ratio, viz., about 11 to 1. They are, however, only lightly braced so that they are fairly flexible in the vertical plane and are, moreover, hinged at the hub to avoid the transmission of gyroscopic forces to the vertical shaft, and for another reason which will be referred to later.

It should be made quite clear that this lifting airscrew is not directly driven by the engine in any way. When taking off from the ground, the engine driving the tractor screw is started up and the lifting screw is set in rotation by hand. At the demonstration on Monday last, this was done by men pulling on a rope wound round a frame carried by the blades, but obviously some more convenient method would be employed in a machine intended for practical service. As the machine is moved forward over the ground by the action of the tractor screw, the speed of rotation of the lifting screw increases owing to the difference in the drags on the blades having their leading and trailing edges, respectively, foremost, until a speed of about 140 R.P.M. is reached, the estimated speed of the machine over the ground being about 30 miles per hour. The pilot can then raise the machine from the ground by means of the elevator. Climbing is effected by the combined action of the lifting and tractor screws, and the machine can be controlled in the vertical plane by the elevator and steered by the rudder similarly to an ordinary aeroplane.

Owing to its rotation in the horizontal plane, the relative velocity of the air and the blades of the airscrew is different on each side of the screw, the lift, of course, being greater on the blade which is advancing into the wind than on that which is moving in the opposite direction. The effect, however, is considerably diminished by the method of connecting the blades to the hub above referred to. The hinges at the roots of the blades permit them to flap up and down as they rotate, and the effect of this is to decrease the effective angle of incidence of the advancing blade and increase that of the receding blade. The high centrifugal forces on the blades also have the effect of bringing the resultant nearer the vertical, and by setting the shaft, previously referred to as vertical, at a slight angle in the opposite direction, the resultant lift is brought almost exactly into the vertical plane. Any discrepancy in this respect which would give the machine a tendency to turn about the longitudinal axis of the fuselage can be controlled by the pilot by means of ailerons, mounted on outriggers braced to the longerons, one of the ailerons being turned up and the other down in flight to keep the machine on an even keel. We were informed, however, that the ailerons, which are clearly visible in both our illustrations, are actually used only to balance the torque of the engine driving the tractor screw. The speed of rotation of the lifting screw, it may be remarked, appears to depend approximately on the weight supported rather than on the air speed of the machine, but doubtless its behavior can be fully explained by the blade-element theory, although this is more difficult to apply than usual on account of the flapping movement of the blades.

During the course of the demonstration flights, the machine rose several times to a height of about 300 feet, making either normal gliding descents or descending very steeply, with the engine cut off in both cases. In the steep descents, the velocity in the line of flight was kept down to about 10 or 12 miles per hour by the action of the lifting screw, which, of course, continued to rotate, so that the landing carriage fitted with oleo shock absorbers was in no way damaged; the run along the ground, in the case of the steep descents, varied from about 1 to 10 yards. The maximum air

speed of the machine, which weighed 2000 pounds, including the pilot, is about 68 miles per hour and its minimum speed 30 miles per hour, although when descending with the engine cut off its velocity in the line of flight is only 10 or 12 miles per hour, as previously stated. We understand that on other occasions the machine has reached a height of 1150 feet. This is certainly a noteworthy performance for a machine of so unconventional a design, especially as it must still be regarded as being in the experimental stage. In any case it is a considerable advance upon the performance of any helicopter with which we are acquainted, and the further development of the type will be watched with interest.—"Engineering" (London), October 23, 1925.

ARC WELDING AS A MANUFACTURING PROCESS.

By H. M. Hobart, M. Inst. C. E. and W. Spraragen.

The arc weld is a very recent addition to the commercial tools available in manufacturing. To a considerable degree its rather wide applications may be said to have come about as a by-product of the war. Delays in replacing broken, worn, and defective castings, and other parts employed in their products or in the machinery of their systems forced upon industrial managers, railroad executives, and others the necessity of repairing such parts by welding. The intensive, successful use of arc welding in making repairs is leading to the gradual adoption of the process in manufacturing as a substitute for riveting where parts have to be fabricated.

These numerous applications of welding are being brought about through economic reasons, among which may be listed: (1) lower cost of the product, (2) increase of production, (3) improvement in the quality of the product, and (4) the overcoming of manufacturing and other difficulties. In fact, each application listed in this article is the result of one or more of these reasons.

Manufacturers who contemplate introducing arc welding into their shops as a fabricating means must give attention to several fundamental considerations. Of these, the more important will now be stated.

DESIGN.

Structures and parts of products which it is intended to weld by the arc process must, of course, be properly designed to conform to the technique of arc welding in order that they shall effectively withstand the stresses which they will meet in service.

It is a wrong basic principle simply to substitute arc welding for riveting in a structure originally designed for riveting. While such a course can sometimes be justified as a temporary expedient, it is not fair to arc welding to judge it from results obtained in such cases. The product in whose manufacture arc-welding processes are to be employed should be designed with full recognition of this circumstance.

As an example of the need for special designs which recognize the fundamentally different basis of the two methods of joining metals it may be noted that whereas for riveted joints it is necessary to overlap the plates or employ straps, the use of arc welding often permits butt joints to be used. This effects a considerable saving beyond that which could be made by retaining the overlap necessary with riveting.

MATERIAL.

The material employed should preferably be such as is known to be most satisfactorily arc welded. Thus while arc welding is often applicable to joining cast-iron parts, the art has not yet been so highly developed for

such a purpose as for welding mild steel. Experience is being rapidly gained in applying the arc-welding process not only to high-carbon steel and to cast iron, but even to special ferrous alloys, and to non-ferrous metals. But the great mass of the most important and satisfactory applications relates to joining mild-steel parts, such as steel castings, steel plates, and rolled structural steel.

STRESSES ALLOWABLE IN WELDED JOINTS.

Adequate care should be taken in determining allowable stresses for the welded joints. In the past, designing engineers who were not familiar with arc welding have been apt to base their determination of the required cross-sectional area of some particular weld on some general limiting value for the tensile strength or shearing strength in pounds per square inch. There are a number of important variables which must be taken into consideration, such as for example: (1) the design of the joint, (2) the chemical composition of the material to be welded, (3) the type of electrode used, (4) the position in which the weld must be made, (5) the general experience of the welder, and (6) the experience of the welder in that particular kind of job.

A steel weld is a steel casting and it is well known that a steel casting will not withstand bending stresses to any such extent as a steel forging of the same tensile strength. This indicates that arc welds withstand tension stresses best. Their ability to withstand shearing stresses comes next in order. The detailed preparation of the parts prior to welding is also very important. The angle of bevel should be suitably proportioned and the surfaces should be free from oil, scale, and rust. It is generally desirable to provide suitably designed jigs or fixtures to hold the pieces in line and in the right plane while being welded.

With a good mild-steel welding wire, such as is described later and with plate having an ultimate tensile strength of 55,000 pounds, the tensile strength of an arc weld made by a reasonably skilled operator may be taken as 40,000 pounds per square inch of weld section. The weld section can usually with advantage be made 10 to 20 per cent greater than the plate section. For fiber stresses in pressure vessels, the safe working stress for arc-welded joints may be taken as about 8000 pounds per square inch of the plate section.

The shearing strength of welds may be taken as 36,000 pounds per square inch. For structural steel work a factor of safety of 4 is suitable. This makes the working tensile stress of arc welds in structural steel 10,000 pounds per square inch and the working shearing stress 9000 pounds per square inch.

Of course, a highly skilled welding operator, even with the materials mentioned, may obtain welds as strong as the original material itself and with special electrodes can produce an arc weld that is stronger than the plate. Although at present there does not exist much data as to the fatigue strength of weld metal, a few tests that have been conducted indicate that one-half of the ultimate tensile strength may be employed as an ultimate endurance limit below which an arc weld will not break for millions of cycles of repetition of stress.

FURTHER CONSIDERATIONS RELATING TO CARBON CONTENT OF THE MATERIAL.

In addition to chemical requirements, there are other properties in steel which are important from a welding viewpoint. With regard to its physical properties, well-made steel is largely dependent upon the amount of carbon in it. This is the first factor to be considered in deciding what chemical analysis should be followed. Other things being equal, the higher the carbon the more difficult is the welding and the less satisfactory is the weld.

Of course, the difference in welding qualities is not of much consequence between 0.10 and 0.20 per cent carbon, but there is usually a marked difference when the carbon content exceeds 0.20 per cent. In arc welding, there is localized heating which affects a high-carbon plate to a greater degree than a low-carbon plate. In plates above 0.20 carbon there is apt to be a hard brittle zone next to the weld. Moreover, in arc welds with low-carbon material, the rigidity of the welded joint transfers any bending or vibratory stresses to the parent plate outside the weld and this is more capable of withstanding such stresses. But in high-carbon material the bending and vibratory stresses will not be so effectively diverted from the welded joint to the parent plate.

WELDING WIRE.

Good welding wire, in so far as chemical composition is concerned, may be insured by following the American Welding Society's specifications.* Chemical composition, however, is not a complete criterion. For example, wide variations in the chemical composition of the deposit are obtained when welding wire of identically the same composition is deposited in the weld by the electric-arc process. The physical structure and the gas content appear to exert considerable influence on the result. However, the phenomena are still far from being understood. A simple test of welding wire additional to the chemical analysis can be made by laying a piece on an iron table and melting it half way through with a gas welding torch. Good welding wire can be distinguished (1) by the ease and smoothness with which it melts, (2) by the evenness and smoothness of the melted surface, and (3) by the lack of sparks. Wire that sparks badly leaves a rough porous surface after welding, and wire that does not flow freely and smoothly is unsuitable for welding. A similar test may be made on the steel to be welded by running the torch flame so as to cut a depression in the steel. If the surface is rough and shows small blowholes it is an indication of unsuitability for welding.

ELECTRIC CURRENT TEST FOR WELDING WIRE.

The melting test may also be conducted with electric welding apparatus by "freezing" one end of the electrode to the work or plate, using the normal welding current for the size of the wire involved. By keeping the wire in direct contact with the plate it will reach a uniform melting condition, but if the current is too high the wire will melt close to the plate only and not present sufficient melted surface for observation. It is therefore necessary to adjust the current so that the wire heats up slowly throughout its entire length.

JIGS AND FIXTURES.

The application of arc welding as a fabricating process can in most instances be made more economical and successful through the use of suitable jigs and fixtures. For quantity production very thorough consideration should be given to this aspect of the process in order that full advantage may be taken of the economies in time and labor which may thus be rendered possible. This point is illustrated in a number of the applications of arc welding described later in this article.

* Bulletin No. 2, American Welding Society.

TESTING THE SKILL OF OPERATORS.

With the proper control and assuming conscientiousness and reasonable skill, the quality of a given product is fairly independent of the operator. On the other hand, he must be experienced in depositing the metal in the way most appropriate for the particular job in hand. The expertness of the operator can in most cases be determined only by physical tests to destruction of test specimens of the work which he is doing. Once the welder has proved himself competent to do a particular job, it rests upon the supervisory and inspecting force to see that the quality of his work is maintained.

On the other hand, great care should be taken not to destroy the initiative of the welder. The welder should not be ignored if he complains about conditions not being right. His complaint should be carefully investigated, as sometimes materials which have been accepted by the purchasing department are nevertheless not suitable for the work in hand.

It should also be pointed out that while some welding operators can do a 100 per cent job on certain work, they are entirely unfitted for other kinds of work where it is not apparent that any greater skill is necessary. For example, a man who through long experience can weld thin steel in a flat position may not be able to do it in an overhead position. Again, he may not be able to weld thick steel plates even in a flat position. In another instance a welder who is an expert on repair work may be entirely unsuitable, without additional training, for production work on steel.

Many experts in the welding field recommend periodical tests of sample welds to check the skill of their welding operators. The testing should be done on the kind of work which the operator is performing. Where facilities for testing welded samples are not available, local arrangements can usually be made with a commercial laboratory on a reasonable basis.

INSPECTION.

Inspection of welded products must differ somewhat according to the nature of the work involved. Certain fundamental principles can, however, be laid down as universally applicable in arc welding.

The importance of proper inspection and supervision cannot be overemphasized. Almost every article and book written on welding prior to the last few years, directly or indirectly, has implied that the quality of the product was 90 per cent dependent on the skill of the welding operator. With the methods generally in vogue up to the last two or three years this statement was probably fairly correct, as in most cases the selection of material, the design, and the technique of the work were left to a considerable extent to the welding operator. Under these circumstances, it would have been even more correct to state that the quality of the job was 100 per cent dependent on the welding operator. When the problems of material, design, and technique are worked out in advance, the welding operator is called upon only to manipulate the electrode to do a particular job. Relieved of these other responsibilities the operator can, after a little training, be depended upon to do the work day in and day out with less percentage of variation in quality than is found in riveting. This, of course, presupposes the existence of a competent inspecting and supervisory force.

TESTING THE FINISHED PRODUCT.

The greatest handicap to welding is stated by many authors to be that we have no satisfactory non-destructive test available to check the quality of the weld. Many conservative engineers argue that after a job is finished it is impossible to know with certainty whether it is a 50 or 100 per cent strength weld. Proponents of welding in turn answer that it is equally

impossible to know the strength and quality of a finished riveted joint, the only difference being that we have grown accustomed to be satisfied with a visual inspection of a riveted joint instead of requiring a test to destruction. This is because riveting has been with us for a long period, the processes have become standardized, and we have learned from experience to place confidence in the product.

This same psychological process is taking place with arc welding. However, whereas the riveting art has been practiced for centuries, we have been using arc welding extensively for a matter of only seven or eight years.

As implied indirectly in previous statements it should be pointed out that the quality of the finished job can be determined in advance, if proper material is used, proper design and technique is employed, and if there is in force a system of regularly checking the skill of the welding operator as to his ability in manipulating the arc.

COST DATA.

A quantity of information now exists on the cost of electric-arc welding and in a number of industries comparisons between riveting and arc welding are available. As pointed out in the introductory remarks, a later portion of this article will be devoted to examples of applications of arc welding and comparisons as to the relative cost and quality of product with previous forms of fabrication, such as riveting.

In one industry, viz., that of shipbuilding, considerable information exists. Tables and charts have been worked out in detail which enable draftsmen and engineers to readily compute the cost of doing a particular job.* (Similar charts based upon experience should be worked out for other industries.) A leading authority in shipbuilding estimates that in the construction of a ship's fittings and in hull construction, savings ranging from 15 per cent to 25 per cent may be effected. The application of welding to ship's fittings is already extensive. But the conservatism of shipbuilding concerns has restricted the use of welding in joining the hull plates, to a few small vessels all made by different firms and the cost of these first applications has, of course, always been high.

SPEED OF ARC WELDING.

The data given in various text books and papers as to the speed of arc welding are very conflicting. There are so many variables as to make any general statement impossible. The following review will throw light on the matter:

There are automatic arc-welding machines which make long seams at several times the speed possible with hand welding. From the data at present available the speed of making such seams in thick plates (say ½-inch thick) is two or three times as great with an automatic machine as with hand welding. With thin material (say ⅛-inch thick) the speed with the automatic machine is from five to ten times as great as by hand. Time is lost in changing electrodes and in adjustments of position. Moreover, the hand operator becomes fatigued while the automatic machine does not.

Material up to and including ¼-inch thick is being successfully butt welded unbevelled on automatic machines. It is not generally realized that the angle of bevel greatly affects the speed. The authors once conducted a series of tests on the hand welding of ½-inch ship plate. Among many other experiments made in the course of these tests, four samples of butt welds each 3 feet long were welded by the same operator. The total angle

* "Applications of Arc Welding to Ship Construction," by E. H. Ewertz, Bulletin No. 6, American Welding Society.

of opening formed by the bevel on the plates being welded in the four cases were 30, 60, 90, and 120 degrees, respectively. The welder deposited metal at about 1.75 pounds per hour in all cases.

It took so much more metal to fill up the 120-degree bevel as to reduce the speed of welding to only 1.07 feet per hour as against 3.22 feet per hour for the 30-degree bevel.*

It is interesting to note that for all four bevels in this investigation, the rate of deposition of electrode was a matter of 0.65 pound per kw. hour at the arc on the basis of a 20-volt drop at the arc. One kilowatt-hour is equal to 860,000 calories; 136,000 calories will melt a pound of mild steel. Therefore if all the energy in the arc were used up in melting electrode material, 6.3 pounds ought to be melted per kilowatt-hour at the arc, *i.e.*, about ten times the amount actually melted. But part of the energy in the arc is required to heat and melt the surface layers to be joined; part may vaporize some of the steel; and part is lost in conduction, radiation, and convection. Even at this late period in the development of arc welding, enough advanced physical research of the fundamental phenomena has not been made to allocate even approximately the 90 per cent of energy not consumed in melting the welding wire.

Apparently, some very exaggerated ideas are common as to the amount of metal which can be deposited per hour in practical high-grade arc welding. Some years ago one of the authors of this article published the statement that for hand welding an operator can in the case of the hull plates of ships deposit about 1.2 pounds of metal per hour when working out in the open (as on ships) amidst many inconveniences, as against about 1.8 pounds per hour in the shop. Confirming this it can be seen from the General Electric Company's report in the latter part of this article that, based on the average for a month, for all sorts of work, the rate of consumption of electrodes was 1 pound per welding operator per hour. This would probably correspond to nearly 2 pounds during actual welding. In melting 13,500 pounds of welding wire some 36,000 kw. hours were consumed, or 0.375 pound per kw. hour from the supply circuit. But this very probably corresponded to some 0.6 to 0.7 pound per kw. hour *at the arc*.

TABLE I

Current Amp.	Volts at Arc	Speed of Welding Ft. Per Hr.
165	15	110
195	16	135
200	18	180
220	18	200

As an experiment, a 12-foot cube tank of ½-inch steel was arc welded in 1918. The total time of actual welding was 165 hours, and this corresponded to using up electrodes at the rate of just 2 pounds per hour. The energy consumed at the arc was about 500 kw. hours, and 334 pounds of welding wire was used, of which 299 pounds was deposited in the welds. There was thus deposited 0.60 pound per kw. hour of energy at the arc. In making this estimate it has been assumed that the current was 167 amperes and that the pressure across the arc was 20 volts.

In place of stating welding speed in feet per hour, one of the authors in 1919 expressed the following view with which we now fully agree:

* Much interesting data on this investigation are given in Pender's "Handbook for Electrical Engineers," in a table on p. 1955.

"The practice heretofore customary of stating the speed of welding in feet per hour has led to endless confusion as it depends on the type of joint, height of weld, and various details. A much better basis is to express the speed of welding in pounds of metal deposited per hour. Data for the pounds of metal deposited per hour are gradually becoming quite definite. The pounds of metal per foot of weld required to be deposited can be readily calculated from the drawings or specifications. With the further available knowledge of the average waste in electrode ends and from other causes, the required amount of electrode material for a given job can be estimated."*

For overhead welding or for welding in difficult places, the speed of deposition is much lower than the value of 1 pound to 2 pounds per hour. This is also true when, instead of mild steel, some other material is to be joined.

We have seen records of butt welding with automatic machines where long seams in ⅜-inch and ½-inch plates have been welded at 15 and 12½ feet per hour with a consumption of 6½ pounds of welding wire per hour. The angle of bevel was 30 degrees and the current from 250 to 300 amperes. It is, however, more usual for automatic welding of butt seams in ½-inch plate to estimate on speeds of only 4 to 8 feet per hour of actual welding, as against somewhat under half these speeds for hand welding.

Very high speeds are obtained when arc welding quite thin material on the automatic machine. The precise values vary with the size of electrode and with the current. As an interesting example of an attempt to study the influence of the current, Table I is included to show the results obtained in butt welding 14 gauge material with ⅛-inch diameter welding wire.

In 1924, the data in Table II were published with the statement that "these figures are actual welding speeds and will be modified on production work by idle time, preparation, etc."

TABLE II

RATE OF TRAVEL (BARE METALLIC ELECTRODE)

Thickness of Plate In.	Feet Per Hour (Butt Weld)		
	Hand Welding	Semi-Automatic Welding	Full-Automatic Welding
1/16	10	20 to 40	75 to 150
⅛	7½	15 to 30	40 to 75
3/16	6¼	12 to 24	25 to 50
¼	5	8 to 16	15 to 40
⅜	2½	5 to 10	6 to 12
½	2	2½ to 5	3½ to 7
¾	1¼	1¾ to 3½	2½ to 5

The speeds in Table III, in terms of pounds of metal deposited per hour, are recommended by the authors for estimates on the arc welding of mild steel by hand.

The speeds in Table III are such as can be maintained by a fairly skilled welder during an eight-hour day and under favorable conditions, such as welding in the shop. The plates to be welded are assumed to be laid flat and at a convenient height for the operator. It is also assumed that the

* "Welding Mild Steel," by H. M. Hobart, "Trans. A. I. E. E.," Feb. 19, 1925, Vol. XXXVIII, p. 63.

edges have been prepared and that the plates are clamped and all ready for welding. For vertical welding, these figures should be multiplied by 0.9. For overhead welding, they should be multiplied by 0.7. For work in the field, where the arc is not protected and the welder subjected to inconveniences, these figures should be multiplied by 0.7.

TABLE III

Thickness of Plates In.	Diameter of Electrodes In.	Approximate Current Amp.	Pounds Deposited Per Hr
1/8	1/8	100	1 4
1/4	5/32	150	1.7
3/8	5/32	160	1 8
3/8	1/4	200	2.2
1/2	5/32	175	2.0
1/2	1/4	225	2.4

EXAMPLES OF ARC WELDING APPLICATIONS

LOCOMOTIVE PARTS.

Arc welding of tubes in boilers is standard practice on several large railroads.

WELDING PIPE LINES FOR OIL, GAS, WATER, STEAM AND DREDGES.

A permanent non-leak joint as well as a cheaper and better job may be obtained by the use of arc welding on all sizes of pipe lines. Miles and miles of welded pipes, varying from small to very large diameter and from very low to very high pressures have been successfully arc welded. They have given satisfaction in service and in a number of instances a saving as high as 25 per cent may be effected over screwed or flanged joints. Angles, ells, tees, and other fittings may be economically welded.

SHIP CONSTRUCTION.

Allusion has already been made to the great field for arc welding, not only in the fittings of ships, but in joining the hull plates. For fittings, the use of arc welding is increasing rapidly. But as a method of joining the hull plates the only applications have been to a few relatively unimportant ships, a list of which is given later in this section of the article. Although the plan of using arc welding instead of riveting for joining the hull plates of ships has been reported upon favorably many times by men of unquestioned authority, wide experience, and mature judgment, the use of riveting for this purpose appears to be so strongly entrenched that it is not likely to be replaced by welding on any considerable scale in the near future. For oil tankers there is with arc-welded hulls the additional advantage of oil tightness. But for steel hulls for any purpose arc welding offers the following advantages: (1) saving of material, (2) saving of labor and (3) increased seaworthiness.

The following are examples of applications in this field:

(1) *Fittings* that are of an important character, such as continuous railing rods, sky-lights, stairs, gratings, ladders, coal chutes, ventilator cowls, stacks, funnels, uptakes, bulkheads (which are not structural parts of the ship), tanks, hand rails, deck houses not covering unprotected openings through weather decks, battery and storage boxes, trap seating, holders for boiler spare parts, drip pans, conning tower hatches, ventilation and exhaust pipes, torpedo doors, periscopes, and ammunition scuttles.

Fittings that are of a minor nature such as lashing pads for torpedo tubes on deck, boat chocks on davits, sockets for hand rails, cable and fixture hangers, cable guards, bucket stowage, pipe supports, berth supports to frames, pipe hangers, leaders on deck for steering cable, chocks under turbines and gear cases, hangers for operating rods, pads for all purposes, bars on escape hatches, cargo batten cleats, protecting angles around manholes, seatings for fire extinguishers, brackets, and waterway bars.

(2) *Clips* for jack rods on galley house, ram rods, stanchions, work benches, battery lockers, stores for paints, oils, provisions, electric galley, torpedo magazines, oil burner rack, oil racks, transoms, gauge boards, lathe and grinding stones, ladders, engine room flooring supports, table, cable hangers, bulkheads, detachable rail stanchions, shelving, and lockers.

(3) *Foundations and Supports* for oscillator, bulkheads, battery hatch, torpedo slides, guns, starting flasks, switchboards, engine and boiler room floorings, discharge manifolds, oil tanks, wash machines, compass, diving, and steering gears.

(4) *Attaching* deck rail stanchions to plating, deck collars around ventilators and funnels, cape rings, galley fixture to plating, bath and other fixtures, cowl supporting rings, engine and boiler room stairs and gratings to plating, cleating to casing, door frames to casings, airtight shoes to bulkhead.

Mention should also be made of the attachment, by arc welding, of anti-submarine bulges to the hulls of battleships. The shell plates of these bulges were also joined by arc welding. The construction of such bulges involves more feet of welding and the use of thicker plates than the hulls of fair sized ships.

At the Norfolk Navy Yard a battle towing target 172 feet in length was completely arc welded in 1920.

UNFIRED PRESSURE VESSELS.

Welding is entirely adequate for the construction of all kinds of pressure vessels. It is confidently expected that the present restrictions in the A. S. M. E. Code in regard to welding will be modified and made more liberal, based upon the scientific evidence now available and soon to be obtained by the Pressure Vessel Committee of the American Bureau of Welding. Tanks designed for 250-pounds pressure have withstood pressures as high as 2400 pounds before bursting. Welding can produce a better and cheaper tank than riveting when properly designed and constructed under approved technique and proper supervision. The test recommended by the American Bureau of Welding for pressure vessels is indicative of the confidence which may be placed in properly constructed vessels. This test calls for hammering a tank when under a hydrostatic pressure of 1½ times the working pressure, after which the pressure is to be raised to three times the working pressure and held there three minutes except in the case of vessels of special construction such as shell and tube brine coolers, condensers, intercoolers, etc., in which case the pressure shall be raised to two times the working pressure. This may be compared with the hydrostatic test of 1½ times the working pressure required by the A. S. M. E. Boiler Code for riveted vessels.

FIRED PRESSURE VESSELS.

Many types of steam and hot-water boilers and fired pressure vessels for various industrial purposes ranging from low pressure to high pressures and high temperatures are now in successful use.

STACKS FOR BUILDINGS.

The Boom Boiler & Welding Co. produces both riveted and welded stacks. It states, however, that a rapidly increasing number of owners are demanding the welded product. Welded construction results in 100 per cent efficiency of joints in a stack as compared to 60 per cent for riveted construction. Breachings and gas and smoke conductors in a stack generally allow leakage when riveted, which spoils the vacuum, hence handicaps the draft. When these parts are welded, the joint is 100 per cent air tight. Joints of a stack that must be subjected to heat strains can be held tight when welded, where they will give if riveted. Moreover, a welded stack has no rivet heads or rough edges to collect soot and soft coal, which form sulphuric acid and eat into the stack.

STEEL FLASKS.

So successful has been the application of arc welding to the building of steel flasks, specially formed shells used in making sand molds in foundries, that one of the largest producers of these has entirely discontinued the manufacturing of the cast or riveted product in favor of the welded steel flasks.

ALL-WELDED STEEL FURNACE.

Welding also gives a leak-proof and strong union in the construction of steel furnaces.—"General Electric Review," December, 1925.

CHARACTERISTICS AND USES OF GROUND GEARS.

By H. F. L. Orcutt.

PART I.

Introduction.—There is at present a demand from engineers for high-duty spur-gearing. This demand will increase, when there is a wider knowledge of the service that it is possible to secure from the straight spur-gear made from hardened steel with the teeth accurately finished. The ground gear is practically a new mechanical product except in motor-car practice, and even in the motor-car gear-box it is adopted by only a portion of the automobile makers. Hence, it may be said that its application or use is so limited that it may be almost regarded as unknown in general engineering, for its virtues, characteristics, and difficulties of production are understood by very few.

Characteristics of Ground Gears.—The two outstanding characteristics of ground gears are (1) extreme accuracy and (2) the possibility of making a straight tooth spur-gear of material with any desirable physical properties. Any alloy steels or heat treatments can be specified, and even with the hardest material an accuracy of tooth finish with limits of errors even smaller, with few exceptions, than are demanded in cylindrical grinding can be obtained. This combination of high grade material and accurate finish of the teeth would be impossible without the grinding wheel. The standard of accuracy which is required in ground gears to make them superior in endurance and noiselessness to cut gears is not generally understood, and it is desirable to say a few words on this subject.

The ordinary commercial gears with the teeth either hobbed, milled, or generated give satisfactory service when they are run at slow speed and when they are not too heavily loaded. The same gears when run at high speeds develop objectionable noises, and unless they are lightly loaded have a short life. Cut gears with straight teeth, as used in the motor-car gear-box, begin to develop objectionable noise even when run at light loads and low speeds.

Experience in motor-car work has demonstrated that to secure quiet running, errors in gear tooth shape, in placement of teeth and indexing, must not exceed 0.0003 inch. It is believed that the task of keeping the errors of 250 teeth (the average number in an ordinary gear-box) uniformly within this limit is, as a mechanical operation, without parallel in production. It certainly cannot be achieved with any cutting tool but the abrasive wheel, and it demands the employment of the most accurate and refined grinding machine. It calls for special experience, highly special measuring instruments and extraordinary inspection and supervision. In fact, the production of really quiet gears requires a special organization and equipment, which the ordinary manufacturer can rarely, if ever, afford to operate. The reasons why this accuracy is essential must now be examined.

In straight spur-gears all tooth action takes place by sliding and all tooth contact occurs on a line. There is, moreover, the additional complication that the load is constantly transferred from one tooth surface to another. This action takes place in an incredibly short time; in many cases 100,000 and even 200,000 times per minute.

The statement that contact takes place on a line cannot, of course, be challenged. The line of contact should be straight and continuous. If it is not, noise is the result. The greater the departure from exact contact the greater the noise. If these conditions are kept in mind, it is not difficult to understand the importance of especially accurate indexing and of smooth tooth finish. The abrasive wheel gives the desirable qualities of finish and accuracy of form, and it is practically impossible to secure them with any other cutting tool. Hence there is in the gear-grinding machine the only instrument which will produce straight teeth gears which will run quietly at high speeds. Moreover, gears with the teeth finished by grinding have special characteristics of great importance to the mechanical engineer. These characteristics are of such a nature that they make possible new developments in power transmission. They will be briefly noted.

The ground gear finished to a degree of accuracy necessary to quiet running possesses full theoretical efficiency in respect to load, life and stresses. Contact can be maintained along the full width of the teeth and with perfect regularity from top to bottom of the working surface. These qualities are not found in any cut gear, and in all gears which are hardened after cutting there is still greater departure from them. In ordinary cut gears the loads are constantly shifted from one part of the tooth to another, and, owing to irregularities of form, stresses are concentrated on high spots. The gear which is hardened or heat treated after cutting has much more strength than the soft steel or cast-iron gear. It is, however, on account of excessive irregularities impracticable for high speeds. It must have very large dimensions, as load is constantly thrown onto short lines of contact.

Gear makers and steel makers have given much attention to the quality and treatment of steel which would have a minimum distortion when heat treated. When the teeth of gears are to be ground, any reasonable distortion need not be feared. This is important, for it permits of the use of material which can be selected solely on account of its physical properties and which can therefore be made of minimum dimensions. In practice the best combination of these desirable qualities is found in a case-hardening steel.

Although the greatest distortion takes place in such steels, these give the longest life and bear the heaviest tooth loads. Hardening distortions vary

with size and shapes of gears and must obviously be kept within certain limits. In small gears as used in the motor-car box, a grinding allowance from 0.003 inch to 0.005 inch is necessary on each side of the teeth, or from 0.006 inch to 0.010 inch on the tooth thickness. On large diameters up to say 30 inches, 0.040 inch on the tooth thickness is necessary or all teeth will not completely finish.

Advantages of Ground Gear Teeth.—With full theoretical efficiency and no limitations respecting choice of material, straight spur-gears can be designed with minimum dimensions. This is valuable in many cases where restrictions are imposed by weight and space. A good example is found in a pair of reduction gears, many of which are in service in a well-known makers' aeroplane engine, with the following dimensions:—

Gear, 12.46 P. D.
Pinion, 5.34 P. D.
Teeth, 3.9326 pitch.
Width of gear, 2.6 inches.
H.P. transmitted, 650.
Load per inch of face, 3450 pounds.
Peripheral speed of gear, 2650 feet per minute.

Any number of gears in motor-car gear-boxes are in use carrying a load of over 3000 pounds per inch of face. As a contrast, many electric railway gears are in use carrying loads of only 1200 pounds per inch of face. Theoretically, the latter might be reduced in width to one-half the size commonly used, that is if properly selected material were employed and the teeth were accurately finished by grinding after hardening.

For railway gears the factor of safety on the gears as now made is 16 on the elastic limit. The question is: Could the factor of safety be reduced to 5 without danger of rupture from shock loads? So far as is known no tests have been made covering this possibility. The gear in the motor car stands enormous shocks and great abuse and rarely fails.

The demand for gears running at very high speeds is rapidly increasing. As a direct, simple, cheap and reliable means of transmitting power, the ordinary spur gear is unequalled for many purposes. When made with straight teeth, however, the ordinary cut gear will only permit of a comparatively slow speed. Not very much is known as to possible speeds attainable with ground gears. So far as tested, however, it seems that quiet running can be secured with high-grade gearing with practically unlimited speeds if the teeth are accurately ground, are not overloaded, and the gears are correctly mounted. A peripheral speed of 3000 feet per minute is commonly in operation with quiet running, and it is reported that there are in use ground gears transmitting considerable power at a peripheral speed of no less than 10,000 feet and even 20,000 feet per minute.

Direct comparative tests have been made which demonstrate that gears accurately and smoothly finished have longer life than those which are inaccurately or roughly finished.

It is suggested that many cases of the so-called "pitting" are directly due to rough or inaccurate finish, the theory being that rough projections on the teeth are pushed by sliding action towards the pitch lines, and near the pitch lines particles of metal are pressed into the surfaces of the teeth, causing these well-known blemishes. Support is given to this theory as pitting action is always found close to the pitch lines of the teeth where the sliding action is least, and rolling action is more nearly approached, and lubrication is less effective than on those parts of the teeth where sliding action is more pronounced. In theory it seems that the ground gear should have the longest life, as it is a fact that in nearly all mechanical action the maximum

wearing qualities exist in those surfaces which most nearly approach mathematical perfection. Actual service tests have shown that railway transmission gears with ground teeth outlast all others.

It is quite possible that there will be distinct advances in gear-tooth service when more gears are designed with a long line of action and with as many teeth in contact as possible. Multiple-tooth contact is, however, not of much practical value in gears when they are finished by ordinary cutters or are hardened after cutting. The slightest imperfections of indexing or of variation in form prevent division of the load, which should in theory be evenly distributed on two or more teeth.

Back-lash, under certain conditions, directly affects the life of gearing. It is much talked about and much misunderstood. To it are attributed erroneously many evils; with its elimination are expected many improvements. It should be mentioned in passing that it does not affect noise, speed, nor to any great extent load capacity. A certain amount is just as necessary to good gear running as clearance is to good spindle running. But the amount of it must be strictly under control, and it has already been seen that variation can be limited to a fraction of a thousandth of an inch by grinding, as compared with variations ten times as great in cut and hardened gears. Uniform teeth forms and uniform back-lash are, of course, natural accompaniments.

Recent tests have been made in America to obtain definite data as to effect of inaccuracy of spacing on the strength of gear teeth at high speeds; it was found that, in a broad way, at pitch velocities of 1000 feet per minute and upwards, gears whose inaccuracies of spacing do not exceed 0.001 inch will carry twice as much load as those having inaccuracies of spacing of 0.006 inch; and strength of gears having inaccuracies of spacing of 0.002 inch is about half-way between the two. These tests were on cut gears, but they are significant and applicable, of course, to ground gears.

Conditions which, as far as known, seem indispensable to the best running results for straight spur-gears, may be briefly summarized:—(1) Accurately formed teeth. (2) Indexing as near perfect as possible. (3) Teeth with sections and area well above those demanded by the load they must carry. (4) Gears mounted so that good tooth contact is maintained. (5) Connecting drives and transmitting shafts, if possible arranged so that oscillations and vibrations are not transferred to the gears.

In respect of (1) and (2) it may be predicted that the ground gear will more closely approach perfection than any other.

PART II.—USES.

The foregoing details must be very carefully considered before the ground gear can be correctly appraised.

The successive stages of evolution of the straight toothed spur-gear are:—(*a*) The wooden cog of the old millwright. (*b*) The foundryman's gear with cast teeth. (*c*) The gear of the machine-tool engineer, either milled, hobbed, or generated. (*d*) The gear hardened after the teeth have been machined. (*e*) And finally the gear with teeth ground after hardening.

It is rather curious that the machine-tool makers have not yet made use of ground gears. Customers will certainly demand better gears. Conditions which the machine-tool makers have to face in the near future will force the pace of adoption. These conditions are of a twofold nature: one is the fact that machine-tool design has never really "caught up" with high-speed steel; the other is founded on the probability that non-ferrous metals—some of the newer bronzes—will very largely replace cast iron. If this occurs, cutting speeds will have to be increased much beyond the present fastest speeds, and in such circumstances the gears as now used will be perfect "howlers," intolerable for noise alone, to say nothing of short life with

costly upkeep and expensive renewals. Ground reduction gears for electric motor drives will probably be extensively made use of by the machine-tool maker, that is, single reduction gears of comparatively small dimensions and run at high speeds, using small power motors, with gear mountings adapted to these conditions.

An important application of ground gears is beginning to interest electrical engineers. The ground gear has many features which make it especially applicable as a direct reduction unit between the motor and axle of electric locomotives. It is simple, smooth running, quiet, will stand very high speeds, can be of small dimensions, has a high factor of safety, and long life. So far as applied it has demonstrated its safety and life and smooth running. It has not, however, been made use of with dimensions and speeds outside of the specifications usual for soft steel. The dimensions stipulated are very much in excess of those which are safe. Why is a factor of safety of 16 necessary? There is much to be done in the way of standard specifications for this gearing. Specifications now usually given are meaningless; as far as fixing limits are concerned they are of no use. The requirement that a gear tooth must be accurately cut with a 14½-degree pressure angle is capable of a wide interpretation. Back-lash, indexing errors and eccentricity are of vital importance in high-duty gearing. Tooth forms are not of much importance as long as they are exact, although design of forms should be considered for each special service and ratio.

A specification for railway traction gears should include (1) smooth finish on teeth, (2) a maximum back-lash of 0.02 inch, (3) a maximum back-lash variation of 0.002 inch, (4) parallelism of teeth, maximum error of 0.001 inch, (5) index errors from tooth to tooth in gear and pinion not exceeding 0.0005 inch.

It is strongly recommended that research work, especially on electric traction gears, be undertaken, as there are practically no data available for this new class of gearing, which in the near future may be extensively used. The hardened gears not ground, now quite extensively in use, both for main line and tramways, are very inaccurate, so inaccurate that it is quite possible they are actually preventing the electric railway engineer from making distinct improvements in his power unit.

So far as is known, the ground gear is not used by any electrical engineer in ordinary engineering speed reductions, so commonly demanded where direct motor drive is employed. There is a very large possible application in this particular direction. The ground gear for motor drives will certainly replace in many instances helical gears, laminated gears, fibre gears and chain drives, in many cases saving in first cost as well as giving an equally quiet and a more lasting service. There are many cases where helical gears are used to secure quiet running where end-thrusts are undesirable. In these cases the straight tooth ground gear may possibly be used as a satisfactory substitute.

For anything except comparatively slow speeds it is now impossible to make use in gearing transmission of the alloy steels with extreme physical properties. When the teeth of gears are ground after heat treatment the engineer has an unlimited choice of metals; he can select that which gives him strength combined with light weight and small sections, or large sections giving the maximum life and withstanding severe shocks and loads. Two extremes may be cited: (1) In the aeroplane where light weight and reliable service are indispensable; (2) in a rolling mill gear, where breakdowns and repairs are costly, and heavy loads and shocks are not to be avoided.

It is quite possible that chain transmissions will be replaced in many instances by ground gears, giving full reliability, quiet running, a saving in first cost, and in weight, and troubles due to stretching chains.

Whether ground gears will be used for turbine transmission very extensively or not it is impossible to say. They have been in service for some time, but only to a limited extent. Gear grinding on large diameters with fine pitches has not been very much developed. There is, however, no reason why it should not be successful. It will certainly be available in the near future with every degree of accuracy now found in small ground gears.

A further use is anticipated in gears where heavy loads are carried and which, made of soft material wear out quickly. In many cases such gearing is replaced at very great expense, chiefly on account of cost incurred through loss of output while repairs are undertaken. If gearing that will last, say, 6 or 9 months, instead of 3 months can be put into such plants, first cost will be of little consideration.

Methods of Grinding.—Two methods of gear tooth grinding are in use: one is adopted by those who have developed the so-called generating machines; the other by those who employ the formed wheel. There are now on the market half-a-dozen different machines of the generating type. In some of them a very large grinding wheel is employed and the width of gear is limited to about 1 inch, as no motion either of the gear being ground or of the wheel parallel with the axis of the gear takes place. In other generating machines there is a reciprocating motion as well as a rolling; on these machines gears of wide face can be ground. In one other type of gear grinder the tooth form is dependent on an involute form from which the gear teeth are "generated." In some generating machines only one side of the tooth is ground, in others both sides of the teeth are finished at once.

In the machines using the formed wheel the entire tooth space is acted upon by the formed wheel revolving while passing through the tooth space. On them, width of gears to be ground is limited to the traverse of a reciprocating slide carrying the formed wheel. So far as is known, none of the formed wheel type of machines are on the market, at least not in the European market.

It has been found necessary to devise and make entirely new measuring instruments for inspecting the teeth of ground gears and for checking index errors. These instruments are absolutely necessary to the production of ground gears. They are of little practical use in the production of cut gears.

PART III.—RESEARCH.

Gear tooth grinding is now on a commercial basis, and gears with accurately ground teeth are available, but not in very large dimensions. The principal development, of course, has been in connection with motor-car transmission gears. Quite a lot has been done on gears for electric traction work. For this class of work gears can now be ground on a commercial basis up to about 3 feet in diameter and 8 inches wide, and 2 diametral pitch. Little or nothing has been done on gears with a finer pitch than 12 diametral. Any material can be ground. The process has been so far tested in motor-car gears as to demonstrate conclusively that the cheapest way to get a quiet gear is to have the teeth ground. Gears for aeroplanes have been successfully ground in considerable numbers.

It should be noted that the future of gear grinding will not be confined to hardened gears. There is now a demand for unhardened straight toothed spur-gears with an accuracy which it is practically impossible to produce uniformly in the cut gear. It is believed that gears finished on the gear tooth grinding machine meet this demand.

Much research work is necessary on straight-toothed gearing. It is important to make tests and experiments with the special object of fixing the limitations of cut gears; these limitations are pretty well known for cast-iron and soft steel. They are not, however, sufficiently well defined on

hardened gears in respect to loads and life. Maximum allowable errors for specific services are not recorded. It is desirable that they should be done, especially for gears made from alloy steel treated to withstand heavy loads and shocks.

Gear tooth grinding is now sufficiently advanced to indicate that further progress in spur gears is almost wholly a matter of accurate finish, and that extensive research on high-duty cut gears is either useless or impossible.

It is difficult to lay down an exact program of research. One of the first things to determine is the maximum practicable accuracy of cut gears, both soft and hard. In this work it will probably be necessary to make use of ground gears finished with predetermined inaccuracies.

When the engineer knows definitely how far he can rely on cut gears, he can decide when he must make use of ground gears. The next step in research work should be made with a view to fixing data for ground gears. Enough facts are known to warrant extensive tests. These tests may be broadly subdivided under: (1) Endurance or life, (2) speeds, (3) noise, (4) loads, (5) accuracy, limits of error and effects of same.

Endurance.—Tests to include gears made from alloy steels, air-hardened, oil-hardened and case-hardened. It is especially important to carry out endurance tests on the case-hardening steels. A special study should be made as to how far a smooth tooth finish affects the life of gears. It is desirable to investigate to what extent accuracy alone does affect endurance.

In connection with endurance tests, separate investigation should be made to determine what materials and treatment are most suitable for spur gears which have to stand severe shock and reversal loads. These tests are especially required for gears used for electric traction purposes, or for services like rolling mills or tube mills. For this latter service long life is also a valuable quality as repairs and renewals are very costly.

Special investigation might be desirable on motor-car gears as to the best material for clashing gears. Gears for this service have to bear heavy running loads but only for short periods, and they must withstand excessive wear from clashing.

Speed Tests.—The limits of accuracy in relation to satisfactory running at high speeds are not definitely fixed. Gear grinders know that certain limits must not be exceeded for a certain service. These limits are finer than those called for in any ordinary mechanical operation, almost as fine as are common in gauge work. The relative importance of the various tooth errors is not thoroughly understood. The various measurements must include tooth forms, indexing, eccentricity, parallelism and tooth depths. A series of tests on each one of these dimensions is necessary to determine how far individually they affect good running. In connection with these tests, lubrication must be investigated. For very high peripheral speeds both the quality of the lubricant and method of application must be considered.

Tooth Areas.—There is very little definitely known as to the tooth areas and loads. Investigation of these details is of extreme importance. Tests to destruction will be necessary. Many motor-car gear-box gears bear loads of over 3000 pounds per inch of face, some 4000 pounds. As these loads are only required for short periods, it is not altogether safe to adopt them for continuous running.

Noise.—It is definitely known that extreme accuracy of tooth form is indispensable to quiet running. The exact relation of accuracy to speed is not known, neither has it been definitely ascertained how far different sorts of inaccuracies are responsible for different noises. Gear grinders are, of course, dealing with these problems daily. The art is sufficiently advanced to give good results, but not enough to satisfy demand for real silence. It is known that beyond certain errors quiet running may not be expected. It is desirable to know how quietly gears will run when they are ultra-accurate, and, of course, how far extreme refinements are commercially attainable.

Minor Tests.—To cover what may be called minor tests includes a study of pitch, angle of tooth pressure, ratios, arc of action and number of teeth in contact. Some curious and as yet unexplained troubles occur in certain gear ratios. The same troubles do not occur in other ratios. These discrepancies may disappear with further perfection in production.

It is suggested that possibly very useful data might be secured by tests of certain ratios in which the design of the teeth varies, for instance with different pressure angles or modified addenda and dedenda, varying these details on a series of tests of gears without varying the ratios.

Efficiency.—In respect to loss of power in transmission possibly there is not very much difference between super-accurate gears and very good cut gears. Still, there must be a difference, and it should be known. With the ground gear it is possible to make comparative tests which have hitherto not been of much value.

Distortion Measurement.—Exact distortion records are certainly desirable, especially for the cut gear. It is necessary to the ground gear as a means of saving production costs in grinding. It is, of course, uneconomical to grind off more stock than is necessary. We should have more definite knowledge as to the differences in distortion between the oil-hardened and the case-hardened gear. In the case of the hardened gear, it is desirable to have the hardened surface as uniform as possible, as it is well known that the hardest portion of gear teeth is on the surface. Distortion tests are not difficult, as they involve only measurements and not much mechanical outfit. It is suggested that these tests may include gears which are accurately ground *before* hardening.

Mounting Gears.—It is apparent that gears with small tooth areas from which long life, high speeds and quiet running are expected must be correctly mounted. Before accurate gears were available, investigations in this important matter were not of very much use. Research in gear mounting is now a necessity. Difficulties with accurate gears are common, and there are many disappointments experienced with those who expect the ground gear to cure all their troubles. Motor-car engineers are just beginning to recognize this problem. In many cases it is found that gear-boxes themselves must be entirely re-designed or details altered before gears will run quietly. In other cases engine and back-axle defects or peculiarities, make themselves objectionable in the gear-box. Connections have to be altered and isolation studied. Certain patterns of gear-box cases seem to be specially designed to magnify vibrations; some are perfect sound boxes. It is well known that the *silent* ball or roller bearing is not yet made. Some are more noisy than others. Limits of workmanship in gear-boxes need defining. Exact center distances are not so important as parallelism of shafts. Clearance in ball-races needs investigation. Reasonance certainly must be a matter of considerable amount of study before the gear-box is completely satisfactory. Marine engineers have been obliged to give serious consideration to gear mounting as well as to the development of exact gear-teeth. It has been found that certain reduction gears will give satisfactory service in one position and will invariably break down if placed in another position between the turbine and the ship propeller.

Bearing Loads and Pressure Angles of Teeth.—Tests made some time ago indicate that the pressure angle of gear teeth does not have very much effect on the bearings, whether the pressure angle is 14½, 20, 22½ or 25 degrees. As pressure angles are increased there is a possible increase in bearing loads of not more than 6 per cent. The tests were made on cut gears. It is certainly desirable to make further tests with ground gears. It is very often necessary to good gear design to increase pressure angles. An alteration of this nature is often objected to on the academic assumption that increased pressure angles increase bearing loads to a degree that is possibly dangerous. —"Engineering" (London), November 27, 1925.

CHROMIUM-NICKEL RUSTLESS STEEL.

By Dr. W. H. Hatfield (Sheffield).

The development in ferro-chromium production of the last two or three decades has permitted the industrial production of very useful steels belonging to the iron-carbon-chromium system. The carbon contents of the early ferro-chromiums were too high to allow of the production of steels with a sufficiently high chromium content along with the low carbon content necessary for resistance to corrosion, combined with real ductility. Ferro-chromiums are now available with a chromium content of towards 70 per cent, with carbon contents of 0.60 per cent, 0.15 per cent, or even lower. Alloys of the iron-carbon-chromium system are therefore now being industrially produced of carbon contents similar to mild and very mild steel, with great success. Amongst these may be mentioned the well-known "stainless steel," used for the manufacture of cutlery, and containing about 13.5 per cent of chromium, with about 0.30 per cent of carbon, a similarly rust-resisting steel for general engineering purposes, of similar composition, malleable steel of similar chromium content but lower carbon, in the neighborhood of 0.15 per cent, and the so-called "stainless iron" of still lower carbon, recently dealt with in these pages by H. S. Primrose. The whole of these stainless or rust-resisting steels are based on a chromium content of between 12 and 14 per cent, and it is this chromium which confers the remarkable resistance to corroding media. If the chromium falls below 12 per cent this resistance is impaired, whilst if it exceeds 14 or 15 per cent complications are introduced, owing to the response of the steels to heat treatment being seriously modified, owing to the austenitic region of the diagram having been entered. It will, therefore, be seen that the chromium content has, for sufficient reason, become standardized, as explained above. The difference in the characteristics of the rust-resisting chromium steels are, therefore, obtained by varying the heat treatment and carbon contents.

It will be appreciated that with a reasonably restricted range of chromium content the range of resistance to corroding media was also limited, and researches were therefore undertaken for the purpose of learning whether, by materially modifying the composition of the steel, the range of resistance could be considerably extended. The investigations which the author had the pleasure of supervision included the determination of the influence of varying contents of chromium, with and without the addition of considerable percentages of other alloying elements. It is not here proposed to detail the whole of these investigations, but rather to describe the very remarkable characteristics of a steel which has been produced belonging to the quaternary iron-carbon-chromium-nickel system. This steel contains approximately 18 per cent of chromium, as against 14 per cent in the stainless steels, 8 per cent nickel, with carbon under 0.20 per cent. The effect of the high chromium content along with the nickel is that the steel is definitely austenitic, and it is as the result of a complete study of the austenitic condition and its response to treatment that the interesting mechanical properties are obtained, together with the extended range of resistance to corroding media.

In view of the importance of the mechanical properties typical values for this chromium-nickel steel, as compared with the plain chromium rustless steels, are given in the table on the following page.

For general engineering purposes, it will be seen that the chromium-nickel steel has similar strength with greater ductility when compared with the plain chromium steel, whilst in its fully softened condition its capacity for plastic deformation is very considerable. The material has, in point of fact, been specially developed, owing to its remarkable ductility, with a view to providing a rustless material which can be cold pressed with ease. It is

being rolled into all the usual forms in which steel is manufactured; it can be hot rolled into rod and tubes, and may be cast successfully into quite intricate shapes. It can readily be wrought and welded; it can also be machined without undue difficulty. It is thus likely to have a considerable range of applications.

The austenitic condition of steels, with the exception of manganese steel, has not hitherto been much exploited. The author considers that these austenitic steels contain great possibilities, and this chromium-nickel steel is a particular instance where this remark applies. In this condition it has a very considerable capacity for plastic deformation, and may be cold drawn

	14% chromium steels.		18% chromium 8% nickel } steel	
	General engineering purposes, hardened and tempered, .30% carbon.	So-called "stainless iron" fully softened.	Condition for engineering purposes.	Fully softened.
Yield point, tons per sq. in. ..	30 0	14.0	28.0	14 5
Max. stress, tons per sq. in. ..	47 0	28 5	51 0	45 0
Elongation, per cent. .. .	26.0	36.0	47 0	62 0
Reduction of area, per cent.	47.0	70.0	35 0	40.0
Brinell number	212	150	216	156
Impact, ft.-lb.	40	90	100	110

into wire and tubes, and effectively cold rolled into extremely thin sheet and strip. The resultant mechanical properties of such products are of great interest, whilst the resistance to corrosion remains satisfactory.

Turning now to the resistance to corrosion, amongst other work undertaken in this field, the influence of varying percentages of nickel and chromium on the solubility of steel in nitric, hydrochloric and sulphuric acids, has been studied in the Brown-Firth research laboratories, and the results published—W. H. Hatfield, Iron and Steel Institute, Vol. II., 1923. The remarkable effect of chromium in increasing the resistance to nitric acid, and that of nickel in increasing the resistance to sulphuric acid, were the outstanding facts emphasized. The increased merit here recorded of the quaternary alloy under consideration follows as a matter of course. The results of investigations made for the purpose of determining the comparative resistance of the plain chromium rust-resisting steels and other metals used in the industrial arts in various corroding media have already been published—see W. H. Hatfield, "The Engineer," December 15th, 1922—and therefore it remains to compare in some measure the increase in resistance obtained by raising the chromium content and by the addition of 8 per cent of nickel, as in the chromium-nickel steel under consideration.

All investigators in this field know how difficult it is to answer such a question as, for instance, "Does stainless steel resist nitric acid?" Everything depends upon the conditions of the test, namely, the concentration of

the acid, the temperature and the general conditions prevailing at the time of the experiment. It is well known that drillings from ordinary structural steel, if immersed in the strongest nitric acid at 15 degrees C. will remain unaffected. If, however, the concentration of the nitric acid is reduced to specific gravity 1.20, violent solution of the drillings takes place with the evolution of nitrous fumes. If plain chromium rustless steel drillings of the recognized composition are immersed in nitric acid of the same specific gravity, *i.e.*, 1.20, at the ordinary temperature, no action takes place. It may, therefore, be claimed that this chromium steel is immune from attack from nitric acid, whereas if the temperature is raised considerably, or if the concentration is reduced below specific gravity 1.033, the steel is attacked. It will thus be seen that resistance to corroding media is only relative.

In regard to the chromium-nickel steel which forms the subject of this article, it is found that the increase in the chromium content and the addition of nickel confer immunity from attack from all concentrations of acid, and at all temperatures up to the boiling point. This interesting resistance to nitric acid obviously fits the material for the manufacture of apparatus used in the manufacture and transit of this acid.

Again, whereas the plain chromium rustless steels are not immune from attack from acetic acid, the chromium-nickel steel is immune from attack from all concentrations at temperatures below the boiling point of the acid. There are a number of other acids and salts where the modified composition confers immunity upon the steel, while the plain chromium steel is attacked, amongst which may be included phosphoric acid, formic acid, tartaric acid, citric acid, lactic acid, calcium chloride and ammonium sulphate.

It is, however, very difficult in a laboratory to reproduce completely the actual working conditions under which a metal is asked to resist a corroding medium, and it is therefore always to be desired that anyone interested in the metal should actually perform experiments under working conditions. It is well within the author's experience that laboratory experiments have shown a metal unsuitable, whereas actual tests under working conditions have proved the contrary, whilst at times laboratory experiments have shown the material to be resistant, yet the working conditions have clearly shown an unsatisfactory response. This is not, of course, at all surprising when one bears in mind the complex conditions frequently existing, and the extreme difficulty of controlling all the variables, which may or may not be of consequence.

With regard to sulphuric and hydrochloric acids, it should be pointed out that this chromium-nickel steel is practically resistant to very dilute sulphuric acid at ordinary temperature, but that it is not resistant as the concentration and temperature are increased. In hydrochloric acid the steel appears to be attacked in all concentrations and at all temperatures above and including the normal.

As regards ordinary corroding influences, *i.e.*, the atmosphere, running water and marine conditions generally, it may be stated that the steel is completely resistant. Samples have been exposed on several coasts where they have been covered by the tide and uncovered at its retreat, and although the exposure has extended over many weeks, no corrosion has taken place. It is therefore to be inferred that here is a steel which does successfully resist the severe corroding influences of marine conditions.

Before leaving the question of resistance to corrosion, the author would like to record the results of some experiments which are of interest from many points of view. It has already been stated that this chromium-nickel steel is not resistant to sulphuric acid at elevated temperatures, but in considering the application of the material to a very important industry, experiments were performed with sulphuric acid plus the addition of nitric acid,

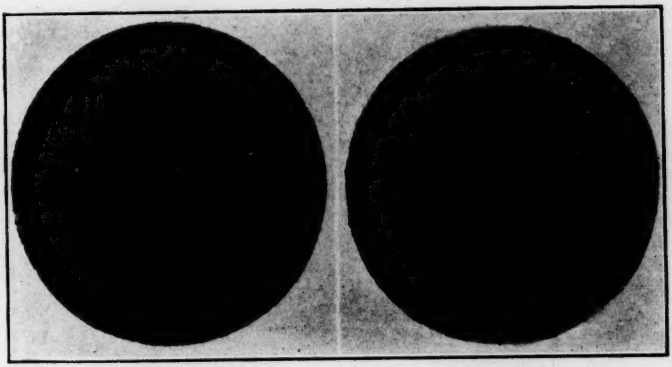


FIG. 1. FIG. 2.

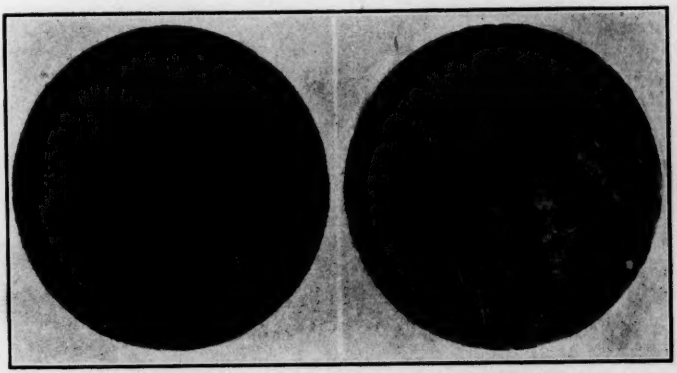


FIG. 3. FIG. 4.

and the somewhat astonishing result was obtained that at temperatures under 100 degrees C. the steel was not affected, whilst at 110 degrees C. the attack was negligible, as shown in the following table:—

18 per cent chromium.....
8 per cent nickel..... } Steel *v.* nitro-sulphuric mixtures

Twenty-four Hours' Test, Temperature 110 Degrees C.

	Loss in weight in grammes per sq. cm.
60 per cent sulphuric acid.....	
20 per cent nitric acid.....	0.0004
20 per cent water.....	
60 per cent sulphuric acid.....	
15 per cent nitric acid.....	0.0007
25 per cent water.....	
60 per cent sulphuric acid.....	
10 per cent nitric acid.....	0.0004
30 per cent water.....	
60 per cent sulphuric acid.....	
5 per cent nitric acid.....	0.0007
35 per cent water.....	

These results led to an important application of this material, both at home and abroad, *i.e.*, in dealing with the "mixed acids" used in the nitration of cellulose.

Turning now to the physical characteristics of the material, the following data may be of interest. The specific gravity ranges from 7.86 to 7.925, according to the heat treatment which the steel has received. The thermal conductivity is 0.029 c.g.s. units, and the specific heat is 0.115 to 0.117 cals. per gramme per degree Centigrade, and the electrical resistivity is 69.00 microhms per centimeter cube.

As regards thermal expansion, the following determinations have been made:—

Mean coefficient 20—100 degrees C. = .000017
20—200 degrees C. = .0000177
20—300 degrees C. = .0000181
20—400 degrees C. = .0000186
20—500 degrees C. = .0000192
20—600 degrees C. = .0000201

During the study of the magnetic permeability of this material very interesting results were obtained. In the fully softened condition the material is practically non-magnetic. Cold working, however, materially affects this property.

A study of the steels under the microscope brings out very well the great difference in constitution and structure. Figure 1 is a micrograph of the ordinary chromium rustless steel for general engineering purposes, at 500 magnifications, and it will be seen that this structure is almost identical with that of the general run of hardened and tempered alloy steels. Figure 2 shows the structure of the so-called "stainless iron," in the fully softened condition, and here the matrix consists of a chromiferous ferrite through which are distributed plates of the double carbide of iron and chromium. Figures 3 and 4 show the structures respectively of the chromium-nickel steel in the condition for general engineering purposes, and also in the fully softened condition. Both these structures illustrate the fact that the steel exists as a solid solution in common with other austenitic steels.

Turning to the manufacturing aspects, it is clear that a steel having the characteristics here described has a considerable future, providing that the cost of manufacture is not too high. Both the chromium steels and this chromium-nickel steel are somewhat handicapped as regards cost, owing to the high percentage of chromium required in their manufacture; but as regards the chromium-nickel steel it is fortunate that, as the steel is in the austenitic condition, its manipulation is comparatively simple, and, with the passing of time, this should be reflected in the economic aspect of the matter. Plates are now being rolled of considerable dimensions up to 20 feet in length by 5 feet in width of varying thicknesses, and the steel is thus available for the construction of fairly large plant. Such plates are rolled from ingots of considerable size, which indicates the comparative ease with which the steel may be rolled and worked in the hot state. Rivets can be manufactured, and, unless in the larger size, may be hammered up cold. In other words, the austenitic condition of this material renders it easily workable, and therefore fits it for a multitude of purposes for which previously non-ferrous materials have been employed.—"The Metallurgist" (Supplement to The Engineer), October 30, 1925.

ALUMINIUM SHIPBUILDING.

It is rather interesting to note that the Japanese Military Traffic Office has decided to experiment with aluminium construction in a small ferry-boat that has been built at the Ujina Dockyard. According to the "Liverpool Journal of Commerce," she is quite a small boat, and it is only the upper works that have been made of aluminium, but it remains to be seen how successful this will prove. Up to now it cannot be said that the results have been encouraging, although they have been going on consistently ever since the early 'nineties. The most interesting of these experiments was carried out by Yarrows', of Poplar, in 1893. At that time the French Navy was tremendously enthusiastic over the small torpedo-boat, just as it is over the submarine today, and the Ministry of Marine determined to experiment with an aluminium torpedo-boat with the idea of getting the highest possible speed in the smallest possible hull. She was quite a small vessel, designed to be hoisted on the torpedo-depôt ship *Foudre*, and the specification stated that she should have a speed of 18¾ knots on a maximum light displacement of 11 tons. The other steel torpedo-boats that were carried by this ship weighed 14 tons apiece, so that the contract was not by any means an easy one.—"Mechanical World," December 25, 1925.

THE BERGIUS PROCESS OF CONVERTING COAL INTO OILS.

So much has been said about the work of Dr. Friedrich Bergius, of Heidelberg, on the liquefaction of coal and the conversion of coal into oils (Berginisation) that a brief account of a lecture on these problems which he recently delivered at Essen ("Zeitschrift des Vereines Deutscher Ingenieure," October 17 and 24) will be of interest. We confine ourselves to Dr. Bergius' own work.

His process, as carried out in the new works at Rheinau, near Mannheim, may be described as a hydrogenation of coal by means of hydrogen at high temperature and pressure. For this purpose he first used the bomb which we illustrated on page 262 of our issue of August 22, 1913. For intermittent working he later used cylindrical autoclaves rotating about their horizontal axes, into which he forced the disintegrated coal, together with some oil as a carrier, and hydrogen at about 100 atmospheres. As the temperature was raised to nearly 500 degrees C., the internal pressure rose to 250 atmospheres and then decreased, because hydrogen was being bound by the carbon. An

experiment yielded, for example, 67 per cent of oil—reckoned on the carbon in the coal, and disregarding ashes, moisture, etc.—16 per cent of residual carbon (insoluble in benzene), some aqueous liquor, and gas. When the hydrogen was replaced by nitrogen the reaction also took place, but the temperature rise and fall were different and the residual carbon was all coked. In some two thousand experiments made with German and foreign coal, the oil yield (reckoned as above) averaged about 50 per cent. Poor grade coal gave relatively more oil than anthracite; the best temperature was 450 degrees to 480 degrees C. The initial pressure should be above 50 atmospheres, and too high a temperature and low pressure favored coke formation. The nature of the liquid carrier, viz., gas tar, coke tar, crude phenols or cresols, did not appear to be important; the liquid facilitated the temperature distribution, but its presence was not essential. The aqueous liquor produced contained some of the oxygen of the coal and air, and, further, most of the nitrogen in the coal, as ammonia; some of the nitrogen entered into the oily compounds; most of the sulphur was eliminated, and could be bound by purposely added iron oxide. The gas, about 20 per cent by weight of the carbon, consisted of methane, ethane, and higher homologues; unsaturated hydrocarbons were not observed.

Industrially an intermittent process, necessitating alternate heating and cooling, recharging, etc., being unprofitable, larger plant was devised for dealing with 300 kg. to 1000 kg. in 24 hours. This consisted of continuous apparatus in which a pump forced a paste of coal ground to 1 mm. grain with 40 per cent of tar (freed of its highly volatile constituents) at 150 atmospheres into the reaction vessel, from which it passed through a coil and a reducing valve into the apparatus for separating liquid and gaseous products. In this plant 1 ton of coal would yield, for example, 445 kg. of oil, 210 kg. of gas, 5 kg. of ammonia, and 350 kg. of residual carbon, which on coking would further give 80 kg. of oil, 240 kg. of coke, and 25 kg. of gas. The whole oil yield would amount to 150 kg. of neutral light hydrocarbons (boiling-point up to 230 degrees C.), 200 kg. of Diesel oil, 60 kg. of lubricants, 80 kg. of fuel oil. The oil was treated as in petroleum works; it contained phenols and cresols, apparently in large proportions. The actual large plant at Rheinau, dealing with some 5 tons daily, has been run for more than a year at an initial pressure of 120 atmospheres, which is to be raised to 150 atmospheres. Iron being weak at the high working temperature, the reaction chamber is indirectly heated by forcing a compressed, preheated, inert gas (nitrogen apparently) through the jacket surrounding the chamber; this gas is afterwards cooled and pumped back into the jacket, an effective heat exchange being provided. The external heating is said to be by a lead bath. Dr. Bergius gave no particulars about these features, nor about the special stuffing-boxes, etc., required. Visitors to the work, we may mention, have spoken highly of the perfection of these devices, which keep the temperature constant for weeks, and of the mechanical arrangements generally.

The process is one of hydrogenation, but not in the ordinary technical sense which depends on catalysts, and therefore requires a very pure hydrogen. Bergius does not use any catalysts and can work with a hydrogen of 80 per cent, such as coke ovens would give; he recovers this hydrogen from the gas of his process, so that coal is practically his only raw material. He can also work in an atmosphere of compressed nitrogen instead of hydrogen, though this is not actually done. Some of the hydrogen required for his reactions is possibly produced by the splitting up of molecules of hydrocarbons formed during the reactions or originally present in the coal. But a supply of hydrogen is essential for his process, and for this and other reasons he recommends the combination of works of his type with gas works. Two or three tons of coal would be required for the production of one ton of oils ready for use.—"Engineering" (London), November 27, 1925.

TIDE POWER.

Every few years public interest centers on some project for the generation of power from ocean tides. Perhaps the most serious recent proposals have been those covering plants at the River Severn in England, the Aber Vrac'h in France, on the Bay of Fundy at Hopewell, New Brunswick, and now at Eastport, Maine, which also lies on the Bay of Fundy at the Canadian frontier. This latest proposal, known as the Passamaquoddy Tidal Power Project and promoted by Dexter P. Cooper, has recently been sanctioned by popular vote in Maine. The waters involved lying partly in Canada, the project awaits approval of the Canadian, New Brunswick and United States governments, as well as the securing of necessary financial backing.

In spite of their paramount importance there is not space for discussion of the economic aspects of tide power in this brief article, which will be confined to a simple outline of some of the scientific principles involved.

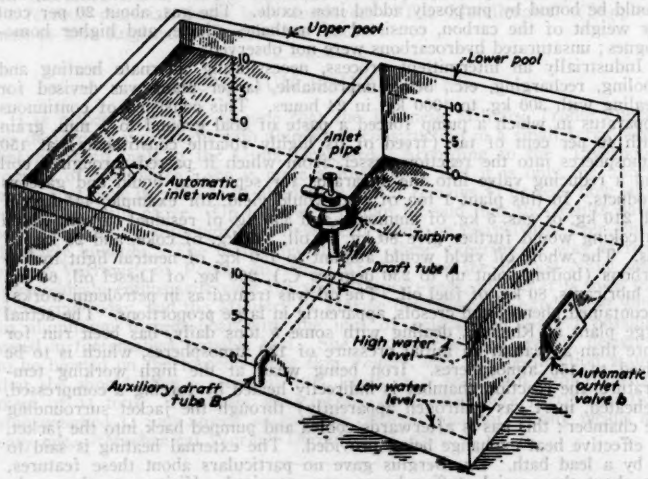


FIG. 1. SMALL MODEL DEMONSTRATES CONTINUOUS POWER PRODUCTION FROM RISE AND FALL OF TIDE.

Ocean tides are caused by the attraction of the sun and moon, principally the latter. The moon attracts all parts of the earth, but pulls hardest on the near side and least on the far side. This attraction tends to raise a hump or wave on the water facing the moon and another hump on the opposite side. The latter is due to the body of the earth being more strongly attracted than the water on the far side.

If there were no celestial motions except the rotation of the earth, these two waves would make the circuit of the earth once every 24 hours, giving two complete tide cycles from high to high in that time. But the moon moves slowly around the earth, rising later each night, so that the apparent rotation of the moon around the earth (representing the combined effects of the earth's rotation and the actual movement of the moon around the earth) requires 24 hours and 50 minutes. In this period there will be two

high tides as the crests of the tidal waves pass the observer and two low tides as the hollows pass him. From one high tide to the next will take about 12½ hours. (Actually 12 hours and 25 minutes.) Normally divided into 6¼ hours of "flood" (rise) and same duration of "ebb" (fall).

The height of the tide—that is, its rise and fall—varies greatly over the earth's surface. For example, it averages 4.4 feet at Governor's Island, New York Harbor, whereas the mean tide is 9.6 feet at the Boston Navy Yard.

Nova Scotia, Canada, is a large peninsula overlapping the whole New Brunswick shore and part of Maine and forming the Bay of Fundy, which is 180 miles long and has an average breadth of 35 miles. Entering the Bay of Fundy along the Maine shore, the tidal rise increases rapidly. At Eastport the mean tide is 18.2 feet, while at Hopewell, N. B., at the head of the bay, the average rise is 38 feet.

The extremely large tides in the Bay of Fundy are due to the piling up of tidal wave between the narrowing shores of the bay. The same effect may be observed when an ordinary wave strikes a funnel-shaped notch in the shore.

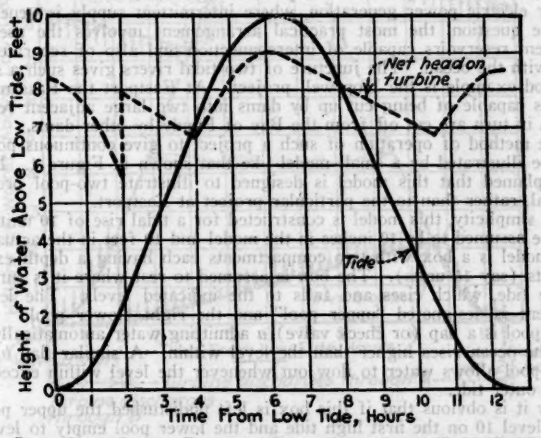


FIG. 2. SOLID LINE SHOWS RISE AND FALL OF A 10-FOOT TIDE; DOTTED LINE GIVES NET HEAD FOR OPERATION AS SHOWN IN FIGURE 3.

The frequent references to "average" or "mean" tides emphasize the fact that the tidal rise and fall at any point varies from day to day. This is due to the fact that the sun produces a secondary tide somewhat smaller than the moon tide. When the sun and moon are together (new moon) or 180 degrees apart (full moon) the two tides add up, giving extreme highs and lows. Such tides are known as "spring" tides.

At the first and third quarters of the moon the two tides oppose each other, giving the low-range "neap" tides. From spring tide to the next neap tide, or the reverse, takes about seven days. At Hopewell the neap tides are about 32 feet, normal tides about 38 feet, and spring tides about 45 feet.

The rise and fall of the tides, when unaffected by the flow of rivers or other disturbing influences, follows what is known in science as a sine

curve, an example of which is shown in Figure 2 (solid line). This represents motion like that of a pendulum—slow near the two extremes and fastest at the middle (that is, at half tide). Note that the tide rises for approximately 6¼ hours and falls for another 6¼ hours, making it about an hour later each day. Thus if we start with dead low tide at midnight the approximate times of high and low tides will be as follows: First day—12 a. m., low tide; 6:15 a. m., high tide; 12:30 p. m., low tide; 6:45 p. m., high tide. Second day—1:00 a. m., low tide; 7:15 a. m., high tide; 1:30 p. m., low tide; 7:45 p. m., high tide.

Practically all serious schemes for using the tide require the presence of large natural reservoirs connected to the ocean by channels sufficiently narrow and shallow to make possible the construction of dams at reasonable cost. With a single reservoir intermittent power can be obtained by impounding the water at high tide and passing it through a wheel to the ocean after the tide has fallen sufficiently.

A small tide mill of this type has been used for sawing lumber at East Boothbay, Maine, for ninety-nine years (see "Power," September 19, 1922). The mill works at odd hours to suit the tide, power being generated by the outflow of water impounded in a pond at high tide.

For electric power generation, where intermittent supply is generally out of the question, the most practical arrangement involves the use of two adjacent reservoirs capable of interconnection and also of separate connection with the ocean. The juncture of two tidal rivers gives such a situation. A good example is the Hopewell project. At Eastport the Passamaquoddy Bay is capable of being cut up by dams into two large adjacent reservoirs, which in turn are cut off from the Bay of Fundy by other dams.

The method of operation of such a project to give continuous power can best be illustrated by a small model like that shown in Figure 1. It should be explained that this model is designed to illustrate two-pool projects in general, rather than to the particular project at Eastport.

For simplicity, this model is constructed for a tidal rise of 10 units, which may be assumed to be 10 inches in the model and 10 feet in the actual ocean. The model is a box with two compartments each having a depth exceeding 10 units (say 15 units). The box is assumed to rest where it is surrounded by the tide, which rises and falls to the indicated levels. The left compartment is designated "upper pool" and the right "lower pool." On the upper pool is a flap (or check valve) *a* admitting water automatically whenever the ocean rises higher than the level within. A similar flap *b*, on the lower pool allows water to flow out whenever the level within exceeds that of the outer tide.

Now it is obvious that if this box is left undisturbed the upper pool will fill to level 10 on the first high tide and the lower pool empty to level 0 on the first low tide.

If water is now allowed to flow from the upper pool through the model turbine to the lower pool, power will be generated. This flow of water will gradually lower the level in the upper pool and raise it in the other, but if the pools are sufficiently large in comparison with the water consumption of the turbine, there will still remain a considerable difference of level up to the time when the water in the upper pool is replenished by the high tide or that in the lower pool is again drained by the low tide. The rise of water in the lower pool may be delayed somewhat by discharging the turbine directly to the ocean for the first hour or two after low tide. This increases the average head available.

One possible method of operation is indicated in Figure 3. This is based on various assumptions that might never be met in practice, but gives an idea of how such problems may be attacked. It is assumed that the water flows through the turbine at a constant rate such as to lower or raise the level in one pool ¼ foot per hour.

The turbine discharges to the ocean for the first two hours after low tide, after which it is switched to the lower pool. Thus, starting with low tide at midnight, the water level in the lower pool stays at zero until 2 a. m. Then it rises steadily to 2¼ feet at 10:30 a. m. At this time the tide has fallen to 2¼ feet, so flap *b* opens and remains open until low tide. The level in pool *b* therefore falls with the tide from 2¼ feet to zero between 10:30 a. m. and 12:30 p. m. The tailwater level of the turbine is the same as the level in the lower pool except from midnight until 2 a. m., when it rises with the tide to about 2¼ feet. Downpointing arrows represent the tailwater level and crosses the lower-pool level.

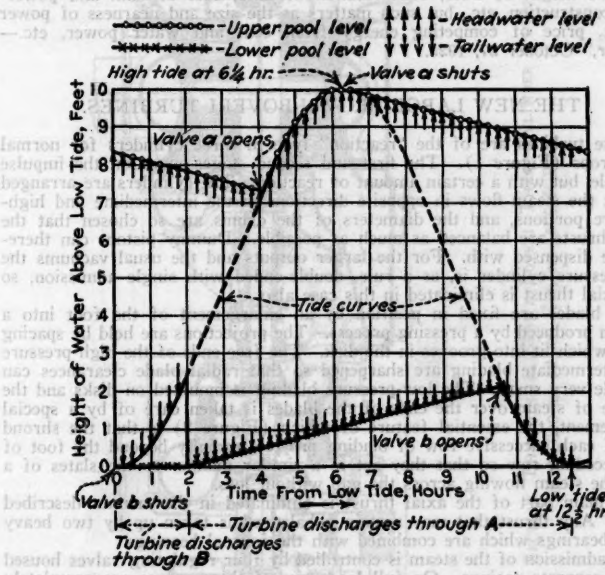


FIG. 3. RELATION OF UPPER AND LOWER POOL LEVELS TO THE TIDE FOR ASSUMED CONDITIONS.

To study headwater and upper-pool conditions (which are identical) start at 6:15 a. m. At that time the tide is high and the pool filled to level 10. As the tide falls flap *a* closes, so that the falling level within is affected only by the flow of water through the turbine. At 12:30 p. m. the upper level will be 8.4 feet. As the diagram does not continue to the right, this level may be carried back to the previous low tide at midnight. From 8.4 feet at midnight the fall continues to 7.4 feet at 4 a. m., when the rising tide reaches the pool level and opens flap *a*. From there until high tide the upper-pool level is the same as the tide level.

The net head available at any time in the tide cycle may be obtained by measuring the distance between the upper and lower levels. This is plotted as the dotted line in Figure 2.

It should be remembered that this cycle would be repeated every 12½ hours, making it an hour later each day.

In practice the demand for power would follow some irregular curve which would fall each day in a different relation to the tide curve, so the actual curve of net head could only approximate roughly that shown in Figure 2.

On the whole the variations in power available from hour to hour and day to day would be much less serious than the seasonal flow variations in the average fresh-water plant.

In closing it should be re-emphasized that a decision regarding the feasibility of developing any particular tide-power project must involve not only the height of tides, conformation of land, cost of dam and power-plant construction, etc., but such matters as the size and nearness of power market, price of competing energy from coal and water power, etc.—"Power," October 27, 1925.

THE NEW LARGE BROWN-BOVERI TURBINES.

These turbines are of the "reaction" type in three cylinders for normal heat drops (Figure 1). The first and second stages work on the impulse principle, but with a certain amount of reaction. The cylinders are arranged so that the steam flows in opposite directions in the intermediate and high-pressure portions, and the diameters of the drums are so chosen that the axial thrusts are balanced as much as possible. Dummy pistons can therefore be dispensed with. For the larger outputs and the usual vacuums the low-pressure cylinder is, as a rule, double-ended with single admission, so that axial thrust is eliminated in this case also.

The blades are fixed in position by an enlargement of the foot into a T-form produced by a pressing process. The projections are held by spacing pieces which fit into grooves in the disk. The free ends of the high-pressure and intermediate blading are sharpened so that radial blade clearances can be made very small. The low-pressure blading is mounted on disks and the leakage of steam over the ends of the blades is taken care of by a special arrangement, the essential feature of which (Figure 2) is that the shroud ring of each successive row of blading projects radially beyond the foot of the preceding row so that they act in a similar manner to the slates of a roof, the steam flowing across the gap without loss.

The main part of the axial thrust is eliminated in the manner described above. Any thrust that may remain unbalanced is taken up by two heavy thrust bearings which are combined with the journal bearings.

The admission of the steam is controlled by four regulating valves housed in two separate casings. On full load, two of these valves are completely open, admitting steam to the complete circumference of the first impulse wheel. On half-load, one valve is open with half admission to the first impulse wheel; on overload, steam flows through the remaining two valves directly into the chambers between the impulse wheels and the reaction blading; on full overload, the steam is not throttled, so that there is no drop in pressure across the impulse stages, which consequently run light. As a result of the use of the impulse principle for the first stages, the distribution of the axial thrust remains unaltered. The regulating valves are operated on the Brown-Boveri oil-pressure, steam-control system.

The superiority of reaction turbines to those working on the impulse principle, both as regards efficiency and constructional simplicity, has in certain quarters occasioned undue emphasis to be placed upon the relatively small blade clearances and the blade-tip losses inseparable from reaction turbines. In fact, some statements which have been made give the impression that if reaction turbines were not made with clearances so small

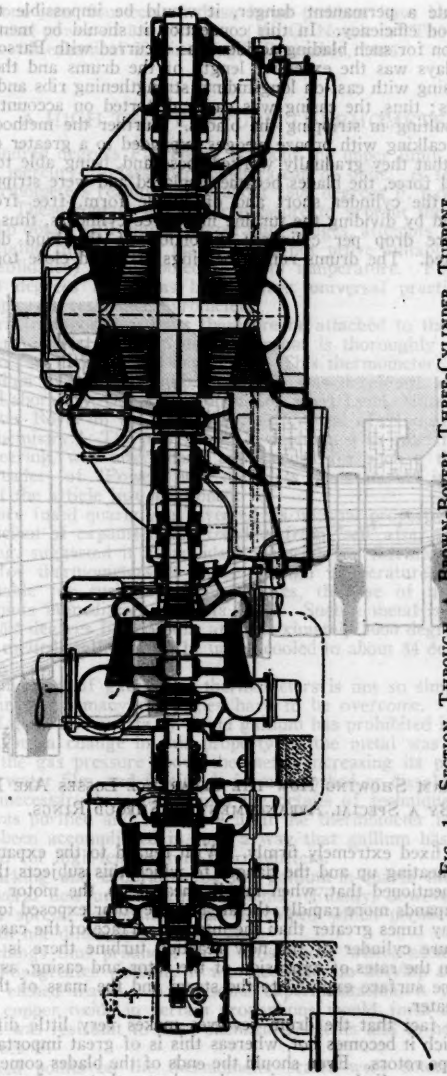


FIG. 1. SECTION THROUGH A BROWN-BOVERI THREE-CYLINDER TURBINE OF 25,000 TO 50,000 KW. AT 1500 R.P.M.

as to constitute a permanent danger, it would be impossible to attain a reasonably good efficiency. In this connection it should be mentioned that the chief reason for such blading accidents as occurred with Parsons turbines in the early days was the excessive length of the drums and the construction of the casing with cast-on longitudinal strengthening ribs and equalizing steam channels; thus, the casing was easily distorted on account of uneven expansion, resulting in stripping the blades. Further the method of fixing the blades by calking with bronze wedges expanded to a greater extent than the drum, so that they gradually worked loose and, being able to withstand the centrifugal force, the blades became displaced and were stripped.

By making the cylinder short and of simple form, free from cast-on projections, and by dividing the turbine into three cylinders, thus decreasing the temperature drop per cylinder; distortion of the kind described is entirely obviated. The drums run in bearings arranged close together, and

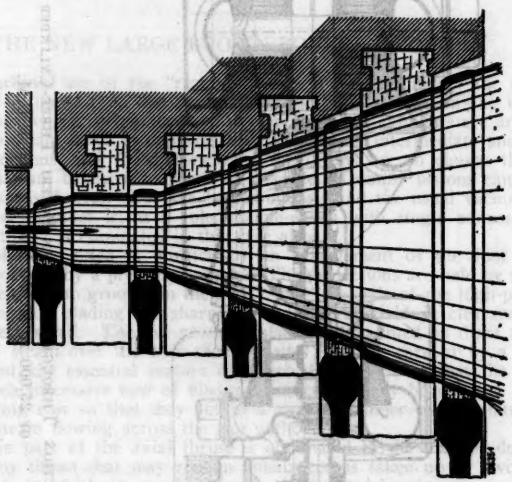


FIG. 2. DIAGRAM SHOWING HOW THE CLEARANCE LOSSES ARE MINIMIZED BY A SPECIAL ARRANGEMENT OF SHROUD RINGS.

the blades are fixed extremely firmly. With regard to the expansion when the turbine is heating up and the danger to which this subjects the blading, it should be mentioned that, when rapidly heated up, the motor of a disk-type turbine expands more rapidly, the area of the rotor exposed to the steam being very many times greater than the internal surface of the casing. With the high-pressure cylinder of the new reaction turbine there is practically no difference in the rates of expansion of the rotor and casing, as both have almost the same surface exposed to the steam and the mass of the rotor is very much greater.

Further, the fact that the drum revolves makes very little difference to the rate at which it becomes hot, whereas this is of great importance in the case of disk-type rotors. Even should the ends of the blades come in contact with the drum or cylinder, experience has shown that no danger is to be feared as the blade tips are sharpened so that they wear down very easily.

Within one year fourteen of these large turbines have been sold, representing a total power of 350,000 kw. (W. G. Noack in "The Brown-Boveri Review.")—"Mechanical Engineering," December, 1925.

A HIGH TEMPERATURE THERMOMETER.

Nine hundred degrees Fahrenheit is about the upper limit of temperature for which mercury-in-glass thermometers can be built. A few mercury-in-quartz thermometers reading up to about 1300 degrees F. have been built in Germany. These contain gas under pressure to prevent the mercury from boiling. To obtain still higher temperature ranges, attempts have been made to use metallic tin, but this material did not prove suitable, as it freezes (that is, solidifies) when cooled to room temperature. For temperatures above 900 degrees F. it has been almost universal practice to use the thermocouples or resistance pyrometers.

Considerable importance must therefore be attached to the announcement that a thermometer has been developed that is thoroughly practicable for temperatures as high as 1800 degrees F. This thermometer, which uses the metal gallium in a quartz bulb and stem, was developed at the Thomson Research Laboratory, General Electric Co., West Lynn, Mass., and described by Sylvester Boyer in the December, 1925, issue of "Industrial and Engineering Chemistry." The power field, in common with other fields of science and engineering, will offer many applications for such a thermometer, so that the readers of "Power" will doubtless be interested in the following abstract of the article just mentioned.

Ever since fused quartz was invented its unusual properties, such as very low co-efficient of expansion and freedom from "set" after alternate cooling and heating, suggested it as an ideal material for thermometers, and particularly for thermometers to measure high temperatures. Mercury not being suitable for such high temperatures, the use of metals with low melting points immediately suggests itself. Such a metal is gallium, which melts at 85.5 degrees F. and boils at approximately 3000 degrees F. Having once been melted, gallium may be under-cooled to about 34 degrees F. before solidifying.

The application of gallium to thermometers is not so simple as it might seem at first, and many difficulties have to be overcome. Heretofore the wetting of glass and quartz by liquid gallium has prohibited its use for thermometers, but a change in this property of the metal was observed when changing the gas pressure above the metal, increasing its purity, removing oxide and water film, and freeing it from absorbed or dissolved gases.

It is unnecessary here to go into the details of technique by which the gallium was purified and kept pure until the thermometer had been sealed. This has been accomplished in such a way that gallium has been made an ideal liquid for high-temperature thermometers.

The marking and calibration of the gallium-in-quartz thermometer presented several new and difficult problems. Finally, however, a protective coating of special composition was developed which permitted etching the marks with hot dilute hydrofluoric acid.

The etchings thus produced were satisfactory where accuracy alone was desired. For greater visibility it was necessary to develop a composition to color the etched marks. After some experimentation, it was found that sand and copper oxide in certain proportions would form a grayish black mark that was resistant to most chemical reagents and to heat-treatment at 1800 degrees F.

The fixed points on the thermometer scales were determined by comparison with a thermocouple as standard or by direct comparison with certain well-known standards for temperatures.

A number of these thermometers have been made during the investigation and have functioned satisfactorily. Others have been loaned to various interested parties for tests.—"Power," December 15, 1925.

PUMPING BILGES WITHOUT BILGE PUMPS.

By Captain Q. B. Newman, U. S. C. G.

The problem of pumping bilges on a steamship is very readily solved by the installation of a reciprocating steam pump, which is small in size and which costs comparatively little to purchase, install, and operate.

On motorships a reciprocating motion is not so readily obtained, and the plunger type of pump becomes more of a problem. It requires an electric

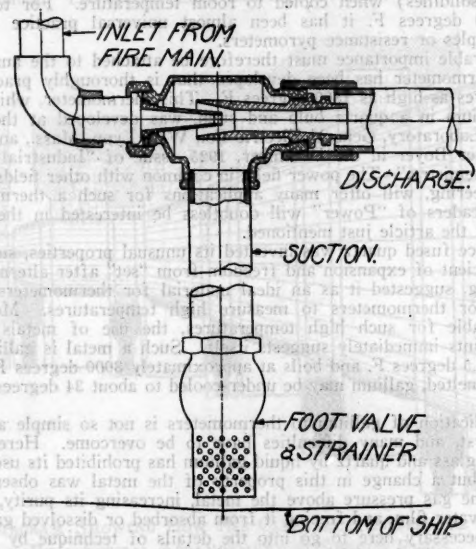


Diagram of Bilge Ejector Adopted on All New Coast Guard Boats.

motor drive which is necessarily of such a speed that it must be geared down before reaching the pump. The pump with its motor, gears, crankshaft and connecting rods is quite expensive, both in first cost and in space occupied.

A vertical sump pump of the centrifugal type is quite satisfactory within the capacity limitations which necessarily exist, but it is frequently not practicable to install a pump of considerable capacity in this fashion.

In drawing plans for the 100-foot Patrol Boats now under construction for the Coast Guard at the Defoe Boat and Motor Works, the problem of pumping bilges appeared to be particularly difficult, and the solution finally arrived at was so simple that it is considered worthy of description.

There has been on the market for a considerable number of years a water jet ejector quite similar in principle and operation to the well-known steam

syphon ejector, commonly found on steamships. The power is furnished by water from the fire main or from any other source where the pressure is 20 pounds per square inch or more. The water passes through an expanding nozzle to the discharge pipe. It creates a partial vacuum in the chamber which encloses the nozzle, and the vacuum thus created is sufficient for a lift of 10 feet or 12 feet. The mixture of "power" water and bilge water passes through the discharge pipe and overboard. There is a combined foot valve and strainer at the lower end of suction pipe.

This type of ejector has been used by the Coast Guard in portable form for wrecking purposes; it will operate at a distance from the ship which is limited only by the length of hose available, and it has been found highly successful for this purpose.

The equipment on a 100-foot Patrol Boat consists of one ejector in each of three compartments. The suction and discharge are 1½ inches and the pressure connection from the fire main is ¾ inch. Control valves are in the engine compartment. The process of pumping bilges consists of starting the fire pump and opening the valve leading to whatever compartment it is desired to pump out.

On Coast Guard Cutter No. 44 (the *New Bear*), the bilge pump equipment will consist of seven ejectors, one in each watertight compartment. The suction is 6 inches, the discharge 6 inches, and the high pressure connection 3 inches. Each ejector has a capacity of 20,000 gallons per hour. There will be two fire pumps on the ship, and when it is necessary to pump bilges, one, or both, of the fire pumps will furnish the water required by the ejectors. In addition there are five ejectors with a capacity of 2000 gallons per hour each, which will be operated from the engine circulating system. For routine pumping of bilges the small ejectors will be more than adequate, and the large ones will be used principally for emergency conditions.

The advantages of the ejector are so marked that there seems scarcely room for argument as to its desirability. It eliminates the bilge piping system, which, with a centrally located bilge pump, is expensive to install, occupies considerable space, and is expensive to keep in effective operating condition. It eliminates the necessity for valves in individual compartments, which in case the compartment should suddenly be flooded, cannot be reached. It has no wearing parts to be destroyed by foreign matter in the water. It can be installed flat on a bulkhead or in bilges, and in either case occupies space which has little, if any, value. It requires no attention or adjustment. It cannot lose its suction or become air-bound. And finally, its cost is very low; the equipment listed above for the *New Bear*, which includes foot valves and strainers, costs about $2,200 for a potential pumping capacity of 150,000 gallons per hour.—"Motorship," November, 1925.

BOOKS RECEIVED.

DIE OELFEUERUNG (OIL-BURNING) BY BRUNO SCHULZ. (IN GERMAN). PUBLISHED BY WILHELM KNAPP, HALLE (SAALE) GERMANY. Price, bound G. M. 12.00; paper G. M. 9.80.

This book of 201 pages of 6½ inches by 9½ inches with 104 figures and many tables is the work of a retired member of the board on "oil firing" in the old German Navy Department. In order to make the work of interest to a larger circle of engineers the experience of the German Navy has been supplemented by compilation of published data from other navies and commercial shipyards and steamship lines have been called upon to contribute. This book covers the subject of oil firing most exhaustively as the names of its seven subdivisions might indicate:

I–Development of oil firing for naval and merchant marine boilers with examples of arrangement on board ship.

II–Oil burners and air regulating apparatus.

III–Auxiliaries, such as oil filters, preheaters, pumps, air chambers and oil measuring devices.

IV–Special designs for oil and coal firing furnaces, oil fired boiler types and peculiarities of boiler installation when using oil.

V–Oil piping and packing, fueling of ships and storing of oil.

VI–Operation of oil fired boilers and their trials on shore and on board, safety measures for prevention of oil fires.

VII–Fuel oil, its varieties and their suitability for oil firing, specifications.

All of the material has been arranged geographically, for German, English, American and all other navies and merchant marines and numerous references in the footnotes give the reader an opportunity to pursue further any particular development. To all those interested in the subject of oil firing of marine boilers the book no doubt will prove a most valuable symposium of existing practice with a large number of cuts to illustrate the text.

E. C. MAGDEBURGER.

JAHRBUCH DER BRENNKRAFTTECHNISCHEN GESELLSCHAFT —VOL. V (1924). (IN GERMAN). PUBLISHED BY WILHELM KNAPP, HALLE (SAALE) GERMANY. Price G. M. 7.80.

This book of 122 pages of 7½ inches by 11 inches with 103 figures, is the official record of the December, 1924, annual meeting of the German Society with the above long name which translated means—"Power from Combustion." The following six papers are included, besides the short minutes of the meeting itself:

1. Internal combustion engines for railroad traction especially the Diesel locomotives, by Prof. H. Nordmann.

2. Ignition process in internal combustion engines, by Dr. Tausz.

3. Diesel engines of large power for marine propulsion, by R. Dreves.

4. Increase in output of internal combustion engines through the use of superchargers and the utilization of exhaust gases for their drive, by W. G. Noack.

5. Fundamentals of high speed oil engines with mechanical injection, by Dr. Buechner.

6. Contributions to the determination of quantitative comparison of various fuels for automotive motors, by Wa. Ostwald.

These papers by representative German engineers and authorities in their line cover the most up-to-date phases of internal combustion engineering. The shipowner has already recognized the Diesel engine as the reliable as well as economical source of power, the railroad executive and even the automobile manufacturer are very much interested in the efforts of oil engine designers throughout the world to adapt it to their requirements. The papers discuss the progress made in Germany towards the solution of these and similar problems. They are of especial interest, beside the designers of internal combustion engines, to all those who seek in oil engines the solution of their power problem and are really worth the trouble that may be encountered in reading them in the original German.

E. C. MAGDEBURGER.

ASSOCIATION NOTES.

A change in the cover of the JOURNAL has been made with this issue incorporating the seal of the Society. The design of four oak leaves shown in the center of the seal was the insignia of the old Engineer Corps of the Navy. This insignia was adopted in 1861 and was worn by the officers of the corps until the engineers and the line were amalgamated.

On the inside cover of the JOURNAL is published a list of the Past Presidents of the Society dating back to 1889 when the first officials were elected.

At a meeting of the Council on December 29, 1925, the officers below were declared elected for the year 1926, following the counting of the ballots cast by the Society:

President:

Rear Admiral M. M. Taylor, U. S. N.

Secretary-Treasurer:

Commander A. M. Charlton, U. S. N.

Members of Council:

Rear Admiral C. W. Dyson, U. S. N.
Captain W. T. Cluverius, U. S. N.
Captain Q. B. Newman, U. S. C. G.
Captain S. M. Robinson, U. S. N.
Lieutenant Commander E. L. Cochrane (CC), U. S. N.
Mr. John F. Metten.
Mr. H. M. Southgate.

ANNUAL DINNER.

At a meeting of the Council of the Society on January 8, 1926, the annual banquet was discussed. The following committee to make arrangements for the dinner was ap-

pointed: Captain W. T. Cluverius, U. S. N., Commander C. S. Root, U. S. C. G., Commander A. M. Charlton, U. S. N., Lieutenant Commander C. S. Gillette, U. S. N. and Lieutenant Commander E. L. Cochrane (CC), U. S. N. The dinner will be held at the New Willard Hotel, Washington, D. C., on March 27, 1926. The usual notice will be sent to all members.

PRIZE ESSAY.

Attention is invited to the special notice on the inside of the back cover in regard to the Prize Essay contest for the year 1926. The prize essay contest was first instituted in 1896 at the instigation of Passed Assistant Engineer Bieg, U. S. Navy.

FINANCIAL STATEMENT.

Following is a statement of the financial transactions of the Society for the year 1925. This statement has been referred to a committee for audit.

STATEMENT OF INCOME AND EXPENDITURES.

Income.

Publication:		
Dues	$6,057.50	
Subscriptions	2,782.25	
Sales	575.70	
Advertisements	4,085.61	
Exchange	1.02	
		$13,502.08
Interest on investments		885.97
		$14,388.05

Expenditures.

Publication:		
Printing	$8,121.65	
Engraving	1,163.39	
Manuscript	1,114.00	
Drafting	14.25	
Postage	135.90	
Commission and discount	253.97	
		$10,803.16

Salaries:				
Secretary-Treasurer		$1,500.00		
Clerk		1,100.00		
			2,600.00	
Banquet, 1925			261.66	
General Expense			146.10	
Current Profit and Loss:				
Charged off for depreciation of furniture		$8.83		
Charged off for bad debts:				
Dues of members	$312.50			
Subscription accounts	1.20			
Sales accounts	6.21			
Old banquet acct. (Carpenter)	5.00			
		324.91		
		$333.74		
Less credit for:				
Received from members for dues previously charged off to P. & L.	$165.00			
Back subscriptions	41.57			
Transfer from "Dues in Advance" of member deceased (J. R. Kyle)	5.00			
Received from Sugar Apparatus Mfg. Co., for interest on advt. account	6.25			
Difference between check book and bank statement (see C. B. 12/31/25)	.10			
Stopping payment on two chks.	3.65			
Recd. payment for advt. direct from advertiser, when charge had been made to agent with commission	9.38	230.95	102.79	13,913.71
Net gain				$474.34

BALANCE STATEMENTS.

Assets December 31, 1925:			
Cash in bank		$2,736.74	
Accounts Receivable:			
Advertisements	$1,940.64		
Dues	449.25		
Sales	35.83		
Subscriptions	89.10		
American Brown Boveri Electric Co.	10.30		
Sundh Electric Co.	6.07		
		2,531.19	
Notes receivable		140.00	

Investments (all bonds):		
Armour Co. of Delaware	$887.50	
Army and Navy Club	1,000.00	
Bethlehem Steel Co.	957.50	
Canadian Northern Ry.	1,115.00	
Hocking Valley Railroad	1,682.50	
International Mercantile Marine	795.00	
New York Steam Corporation	927.50	
Ohio Power Company	860.00	
Penna. Power & Light Company	882.50	
Swift & Company	917.50	
Washington Ry. & Elec. Company	4,230.00	
		14,255.00
Furniture		80.00
		$19,742.93
Liabilities:		
Dues paid in advance	$169.81	
Subscriptions paid in advance	954.50	
		1,124.31
		$18,618.62
Assets January 1, 1925		18,144.28
Net gain		$474.34

By order of the Council, the securities shown in the foregoing statement are carried at the market price as of 31 December, 1923. The market value of these securities on 31 December, 1925, was considerably above the book value.

MEMBERSHIP.

The following members have joined the Society or have been transferred from one grade to another since the publication of the last JOURNAL:

NAVAL MEMBERS.

Townsend, Guy D., Lieutenant, U. S. N.
Whitmer, Donald T., Lieutenant, U. S. N.

CIVIL MEMBERS.

J. J. Fagan, Chief Engineer, S. S. *Leviathan*, Leonardo, N. J.
Henry H. Hower, 818 Auburn Avenue, Buffalo, N. Y.

ASSOCIATE MEMBERS.

W. L. Bunker, 530 Senator St., Brooklyn, N. Y.

H. A. Gosnell, The Nassau Club, Princeton, N. J.

J. A. Sjostrom, Kockums Mek, Verkstads A. G. Malmo, Sweden.

TRANSFERRED FROM ASSOCIATE TO CIVIL MEMBERSHIP.

Howard J. Ball, Bureau of Engineering, Navy Dept.

John H. Schombert, Bureau of Engineering, Navy Dept.